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THE JOURNAL OF THE SOCIETY OF AUTOMOTIVE ENGINEERS

MARCH 1926

SOCIETY OF AUTOMOTIVE ENGINEERS INC.
29 WEST 39TH STREET NEW YORK

The farther a car spring is bent, the greater becomes its shoot-back force — just the same as a bow and arrow, the car passengers taking the place of the arrow.

There is nothing clever about a Watson Stabilator — it's just common logic that to properly control a shoot-back force which varies, the resistance must vary likewise.

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STABILATORS
They hold in proportion to the need

THE JOURNAL OF THE SOCIETY OF AUTOMOTIVE ENGINEERS

Vol. XVIII March, 1926 No. 3

Chronicle and Comment

Tractor Meeting Postponed

AT the suggestion of the American Society of Agricultural Engineers, with which the Society planned cooperation in a Tractor Meeting to be held at Chicago on March 25 and 26, this event has been indefinitely postponed. Further announcement will appear in later issues of THE JOURNAL.

Summer Meeting Plans Progress

FINAL arrangements have been made at French Lick Springs, Ind., for Summer Meeting accommodations for approximately 1000 members and guests of the Society during the period June 1 to 4 inclusive. The facilities at this popular resort are such as to assure a most pleasurable and profitable visit for all who find it possible to attend the Summer Meeting. Many will no doubt wish to stop over at Indianapolis to witness the races that will take place there on May 31, the day before the meeting opens. They should at once make their seat reservations by addressing Indianapolis Motor Speedway Co., Indianapolis.

It is practically certain that the ever-active and hospitable Indiana Section will hold, on the evening of Sunday, May 30, one of its peerless banquets for the entertainment of visiting engineers.

Watch for further information concerning these events in forthcoming issues of THE JOURNAL!

Sections Offer Great Advantages

SOCIETY members have every reason to be proud of the records that are being made by the local Sections. In practically every instance the officers, committeemen and members have demonstrated great wisdom and efficiency in the administration of Section affairs throughout the year. The scope and caliber of the several programs of meetings in the last month typically reveal the truth of this statement.

Milwaukee heard A. W. Herrington discuss our traffic problems; Washington listened to Dr. G. W. Vinal, one of the Country's leading experts on storage-batteries; Pennsylvania received valuable data from A. E. Hutt on serviceing problems; Dayton had the pleasure of hearing a very elaborate paper on sleeve-valve engines by P. M. Heldt; Indiana staged a "three-ring circus" on oil-rectifiers and crankcase-oil dilution; Southern California kept abreast the times with a study of the cooling problem; Cleveland took account of stock and glimpsed into the future of automobile design and construction; Metropolitan considered the various angles of motorcoach operation; Northern California enjoyed an illustrated lecture on radio; Chicago found interest in the engine-fuel question; Buffalo heard Arthur Nutt outline the progress that has been made in airplane-engine design; and Detroit made a decided "scoop" by scheduling Lieut. Cyrus Bettis, Pulitzer Trophy winner, to discuss airplane-engine maintenance.

The advantages offered to Section members by these valuable meetings can hardly be overestimated.

1926 Roster

THE 1926 Roster is being mailed to the 2700 members who ordered copies. As heretofore, the main divisions of this printed list of the membership of the Society are the alphabetical register of members; the list of companies, with names of members associated therewith; and the register of members arranged according to their residence. In addition, members connected with the Army, the Navy, United States Government bureaus, and schools and universities are listed separately. The book contains also uptodate lists of the officers, committeemen and representatives of the Society and of officers of the Sections nominated for service during the coming season. Among the miscellaneous bits of information given is a tabulation of the number of members residing in the various States of this Country. Over 4600 members are located in 11 States. In numerical order these are

Michigan	1,023
New York	964
Ohio	598
Illinois	454
Pennsylvania	402
Indiana	242
California	224
New Jersey	213
Massachusetts	198
Wisconsin	163
Connecticut	127
Total for 11 States	4,608

The other members residing in this Country are located in 34 other States. In addition, 107 Affiliate Member Representatives and 297 Enrolled Students

resident in this Country and abroad are on the rolls of the Society. Two hundred and sixty-nine members have their places of business outside of this Country. The total number on the rolls of the Society as of Feb 15, 1926, which is the date for which all of the figures are given above, is 5799.

An Uptodate Section Constitution Planned

NO formal amendment to the standard Section Constitution has been made since 1920. From time to time in recent years the Council has approved various recommendations of the Sections Committee which involve amendment of the standard Section Constitution and By-Laws. These recommendations, together with certain changes involved in following the new method of collecting the Section dues through the Society office, have necessitated a general revamping of the document.

The Sections Committee has prepared a redraft of the Sections Constitution, including certain changes that are deemed advisable. The proposed revision, copies of which have been forwarded to each Section, represent some very intensive work on the part of the Sections Committee, and it is believed that the document in its revised form is uptodate and suitable for adoption by each of the Sections as a standard Constitution.

The Sections Committee recommends that the amendments, after a thorough discussion by the various Governing Committees, be presented to each Section at its March meeting. After being duly seconded, the amendments can then be brought up for discussion at a subsequent business meeting, which will doubtless be the April meeting of each Section.

The detail changes submitted for discussion will be found on p. 281 of this issue of THE JOURNAL. The changes suggested in SC 6, SC 7, SC 9, and the first paragraph of SC 11 were approved some time ago by the Council upon recommendation of the Sections Committee. The amendment to the second paragraph of SC 11 was suggested so that a member whose dues remain unpaid for a whole year shall automatically have his name taken from the Section roll, without the need for any action by the Governing Committee as has hitherto been the case.

The change in SC 17 was suggested in the interest of simplification.

SB 5, as it stands, provides that a member who has served as Chairman for one year shall not be eligible to immediate reelection to the same office. The proposed amendment would make him ineligible also for immediate election to the office of Vice-Chairman at the end of his term as Chairman. It is realized that this proposed change is contrary to the practice of a number of Sections in which a retiring Chairman is customarily elected to the office of Vice-Chairman. Several considerations influenced the Sections Committee to propose this change. For one thing, the possibility has been pointed out of a new Chairman's moving out of the Section territory shortly after his election, with the consequence that the former Chairman, being Vice-Chairman, would succeed him and would therefore serve as Chairman for two successive terms. It has also been pointed out that the vice-chairmanship is a good office in which to try out new material with the possibility that the Vice-Chairman who gives promise of being a good Chairman may be rewarded for good work by election to that office. It also happens sometimes that a Chairman, after a very busy year in office, feels like stepping aside and letting others take their turn at doing the work, in which case the Governing Committee and the Section lose by having an inactive Vice-Chairman.

The second proposed change in SB 5 is, like the present form of SC 6, SC 7, SC 9, and the first paragraph of SC 11, the result of Council action upon a recommendation of the Sections Committee.

The omission of SB 9 was suggested because it was believed that, under the provisions of the revised SC 9, it has become unnecessary.

It is thought tht the suggested changes in SB 10, SB 11, SB 13, and SB 18 require no comment or explanation.

Certain revisions in the Rules have also been suggested. The detail changes in these are not given with the detail changes in the Section Constitution and By-Laws on p. 281, because it is believed that they can be adequately explained here.

SR 2 as now written provides that manuscripts should be in the hands of the Governing Committee at least 15 days before the date at which they are to be presented. The suggestion is made that the number, 15, be replaced by the number 30, on the ground that it is better to have manuscripts ready 30 days before a meeting, if possible. It is to be noted from the wording of the rule that this is not a matter of compulsion but merely an expression of advisability.

Under SR 5, the right has hitherto been reserved to edit accepted manuscripts of authors as to style, form and substance, and this rule has required that papers shall be read only in the form approved by the Governing Committee. It is proposed to make the latter provision state that *edited* papers shall be read only in the form approved by the Governing Committee.

According to the present reading of SR 10, authors of papers printed by the Society will upon request be furnished with 20 complimentary copies of the same in separate pamphlet or JOURNAL form. It is proposed to insert before the quantity "20" the expression "not more than," and to delete the words "separate pamphlet or." It is believed that the reasons for these changes are readily apparent.

Discussion at meetings is the theme of the rules now designated as SR 12, SR 13 and SR 14. The proposed revision will combine these rules so that they will be numbered SR 12 and SR 13. The last sentence of the rule that will be called SR 12 says: "Comments upon previous discussion will be out of order." It is proposed to delete this sentence as it is considered that this is a matter that may well be left to the discretion of the Chairman.

The rule that will be numbered SR 13 has hitherto contained the provision that in all cases order of preference will be given to written over oral discussion as contributing to the greater clarity, pertinence and value of the record. It is proposed to delete the sentence containing this provision, as the Sections Committee felt that this is another matter that should be left to the discretion of the Chairman, as it sometimes happens that written discussion is not of greater clarity, pertinence and value than oral discussion.

TRACTOR MEETING POSTPONED

Meetings Committee Decides To Join with Agricultural Engineers in Postponement

It is not often that the Society announces a meeting date only to report postponement soon thereafter. But unfortunately it has seemed advisable to "call off" for the time being the Tractor Meeting that was scheduled cooperatively by the Society and the American Society of Agricultural Engineers for Chicago on March 25 and 26. Further announcements regarding the meeting will be published in due course in these columns.

PRODUCTION MEN ACTIVE

Chairman Hill Appoints V. P. Rumely To Head Production Meeting Committee

V. P. Rumely, of the Hudson Motor Car Co., has been appointed chairman of the Production Meeting Committee by Chairman L. Clayton Hill of the Meetings Committee. Mr. Rumely has already been able to give much thought to the tentative program that has been the subject of discussion by a number of those who are especially active and well known in the field of automotive production. A detailed schedule of topics and speakers should develop well in advance of the meeting next fall. Additional information will be found from time to time in these columns.

FRENCH LICK SPRINGS IN JUNE

Popular Resort To Welcome Hundreds on Day Following Indianapolis Races

Not by mere chance was French Lick Springs, Ind., selected as the site of the 1926 Summer Meeting on June 1 to 4, inclusive; it was the choice of a host of members who expressed their wishes by vote. And there were many good reasons why the appeal of French Lick was so strong. No better environment could be imagined, facilities for out-of-door sports and recreation are the best, housing accommodations at the French Lick Springs Hotel are adequate and entirely acceptable, the cuisine is excellent and practically every other feature fully measures up to the highest standards. This is especially significant since the requirements for the Society's Semi-Annual Meeting are not easily fulfilled.

Technical Program in Making

A goodly number of engineering topics have been chosen by the Meetings Committee as subjects for discussion at the technical sessions. The formulating of an attractive program is already well under way, and it is not too early to predict that there will be a number of very stimulating, not to say sensational, revelations at the sessions. The well established practice of including many exhibits and demonstrations will be followed.

Sports Facilities Unequaled

With two splendid 18-hole golf courses, tennis courts and other sports material available and ready for use and with a generous portion of the country's most beautiful open space awaiting occupancy, French Lick Springs and vicinity will extend a cordial welcome to our members. A number of "sure-fire" pleasant surprises are in store for those who enjoy diversion, either mild or strenuous.

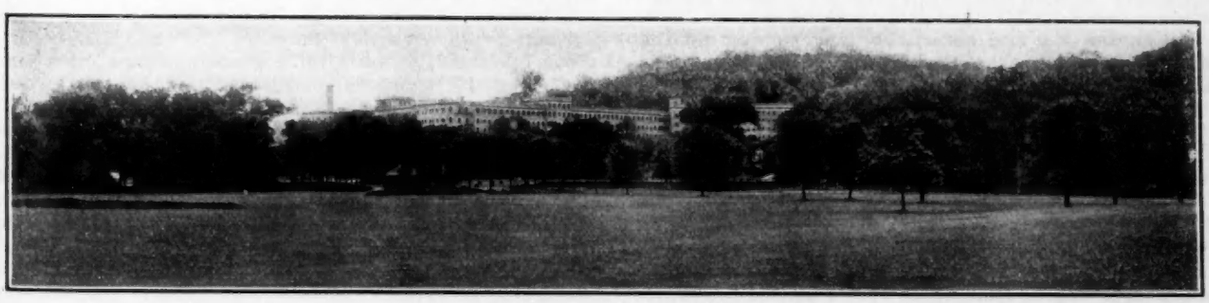
Many will wish to drive their cars over the highways that are said to be first-class in any variety of weather; according to reliable sources of information there is no possibility of a repetition of the difficulties experienced last summer on the West Virginia roads that suffered from the rain.

The Races en Route

Many of those who attend the Summer meeting will wish to stop at Indianapolis to witness the 500-mile classic that will be held on May 31, the day before the curtain rises at French Lick. Reservations for seating accommodations at this event are known to be disappearing rapidly; so those interested are advised to start negotiations immediately with Indianapolis Motor Speedway Co., Indianapolis. In all probability the Indiana Section will hold one of its famous banquets on the evening of May 30 in honor of visiting engineers.

Make Your Plans Now!

It is not too early to mark the calendar to show that you plan to spend the first four days of June with your fellow engineers at French Lick Springs, Ind.! Do it now! June 1, 2, 3 and 4 should appear in red!



A Portion of the Grounds at French Lick Springs, Ind.
The Summer Meeting Will Be Held Here on June 1 to 4, Inclusive, Following the Indianapolis Races

Further information will be issued from time to time and applications for hotel accommodations will be circulated well in advance of the meeting date.

MOTORCOACH DEVELOPMENTS

Service Requirements and Their Influences Discussed at Metropolitan Meeting

That members of the Metropolitan Section are keenly interested in motorcoach design and operation was evident at the monthly meeting of the Section on Feb. 18 at the Building Trades Club, New York City, when the evening was spent in hearing four addresses on the subject and in a spirited discussion afterward. The technical meeting was preceded by a supper which was largely attended by the members and guests and at which a number of cast-steel toy motorcoaches about 15 in. long were presented to those in attendance, with the compliments of the Fageol Motors Co., as appropriate souvenirs of the occasion.

Design Requirements for City Service

City service, said Dean J. Locke, staff engineer of the Public Service Railway Co. of New Jersey, requires greater head-room, wider doors and aisles and fewer cross-seats than interurban service, and a low floor-level to facili the head-room is from 6 ft. 4 in. to 6 ft. 6 in. A floor from 22 to 26 in. from the ground requires only one step up from tate quick loading and unloading. To accommodate standees the curb and two from the ground. Some operators are considering the use of a rear exit to induce passengers to move toward the rear and avoid congestion at the front. Large passenger-loads and slow speed create problems in heating, lighting and ventilating that call for the utmost skill of the designer, he said. The use of a three-part windshield with hinged upper and lower sections, a louver in the side panel to the left of the driver and four large suction-type ventilators in the roof have been helpful in ventilation. A minimum lighting of 5 cp. per passenger-seat is required by the New Jersey Public Utilities Commission but the railway company's newest equipment will have ten 21-cp. body lights and provide 6.8 cp. per seat.

Over-all width of the latest motorcoaches has been reduced, said Mr. Locke, from the former 96 in. to 90 in. to permit of greater speed in heavy traffic and avoid side-swiping. Heavy passenger-loads call for rugged frame, clutch, transmission, and rear axle, while the need for rapid acceleration demands a powerful snappy engine. Reliability and low cost of maintenance are essential to successful operation, hence the chassis must be designed for continuous and unfailing service and to eliminate or minimize vibration and shocks. Posts and side panels should be more numerous than in the body of the interurban motorcoach so as to localize damage from collisions, and they should be easily and quickly replaceable.

Engine Requirements of Interurban Motorcoach

After commenting upon the great demands made upon the engine in the interurban motorcoach, L. P. Kalb, of the Continental Motors Corporation, gave formulas for fuel consumption, acceleration and hill-climbing ability, gear-ratio, and engine size and horsepower and pointed out that the logical engine for the interurban vehicle is the six-cylinder type and that it should have sufficient piston displacement to develop the torque required for good acceleration with a gear-ratio that will not require excessive engine-speed at maximum vehicle speed. For good engine life, the sustained engine-speed should not exceed 1800 r.p.m., and for a sustained vehicle speed of 45 m.p.h. the gear-ratio should be approximately 4¼ to 1 with 36-in. tires. It is desirable to govern the engine speed, and a governor is more necessary on a six-cylinder engine than on a four-cylinder engine. A large engine running at 1800 r.p.m. is preferable to a small high-speed engine, but extreme care must be exercised, he said, to keep the weight down. The speaker then discussed in order the details of a suitable powerplant for the interurban motorcoach.

Problems of Intercity Service

Successful intercity motorcoach operation calls for the exercise of good judgment in the selection of vehicles, routes and drivers; a knowledge of human nature and habits; a careful planning of running schedules and their maintenance; and the wise rendering of safe, quick and comfortable service to patrons. Thus may be summarized the many practical suggestions and observations made by Alexander Shapiro, vice president of the Washington Rapid Transit Co., who came from the National capital to deliver an address on the problems of intercity operation.

Coaches must rival private automobiles in appearance, comfort and speed to sell themselves to the public, said Mr. Shapiro, yet they must be dependable and economical in operation and maintenance. Too large a vehicle means excessive operating-cost and reduced frequency of service, and over-long vehicle life means inability to provide new and improved equipment as it is developed. The selection of drivers presents a problem, as they must be entrusted with vehicles costing from $8,000 to $10,000 and must have the intelligence, initiative, disposition, and honesty to meet any situation without aid. Potential traffic must be studied before deciding on a route, which should preferably connect with other lines of transportation and pass by the best and easiest road through intermediate settlements not on rail lines. The schedule should be arranged so that the motorcoaches will meet trains or motor coaches on connecting lines. Stop-over places should be arranged for at 2-hr. running intervals to allow passengers a chance to stretch and secure refreshments. Once fixed, the schedule should not be changed and coaches should never run ahead of time. Moderate, safe speed should be maintained. Dependability of service is a requisite to success, said Mr. Shapiro, and drivers should become familiar with the traveling habits of their patrons and be on the lookout for them along the route.

Service Planned for Westchester County

Comprehensive plans for a complete motorcoach service throughout Westchester County, N. Y., just above the Bronx, were outlined by A. E. Hutt, president of the Westchester County Bus Co., who concluded his talk by exhibiting several lantern slides of maps of the County showing the location of the railroads and electric lines and the distribution of cities and villages. Two slides showed drawings of a suggested type of low double-deck coach believed by the speaker to be the most suitable for the development of scenic routes through the comprehensive system of parkway boulevards. This design affords full head-room over the center aisle, with a low roof-section over the lower cross-seats. A second level of cross-seats is placed on the low portions of the roof and the raised central portion is used as a second-deck aisle, with a step-down to the seats. The over-all height of such a body, said Mr. Hutt, will be 9 ft. 4 in. Mounted on a mechanical-drive chassis with a frame 24 in. above the ground, this would give an over-all height of 11 ft. On a low gasoline-electric chassis, the body need not stand higher than 10 ft. Upper-deck passengers can be protected from rain and snow by drop sashes on the outer side and storm curtains along the aisle.

The complete motorcoach system as laid out for the County covers about 300 miles of routes, which constitute about one-third of the total highway mileage, and connect all of the 23 incorporated villages and the 17 townships with the county seat of White Plains.

Mr. Kost, of the Firestone Tire & Rubber Co., commented upon the importance of the tire question and stated that his company has under way a tire-service plan that will mean very much to motorcoach operators and that the motorcoach builders have tire equipment plans that will also mean much to operators. E. W. Templin, of the Six Wheel Co., remarked that balloon tires are coming into the picture for interurban operations and that the tandem rear-wheel arrangement is proper for the application of the tires and is proving economical.

Three Papers at March Meeting

The March meeting of the Section is scheduled for the 18th and there are to be three papers on the general subject of Fuels and Lubricants. A. J. Cramer, of the Bureau of Mines, is to deal with the Fuel Situation from the Producer's Standpoint; S. W. Sparrow, of the Bureau of Standards, will tell of Possible Improvements by the Automotive Manufacturer, and J. Willard Lord, service manager of the Harrolds Motor Car Co., is on the program to tell What the Operator Can Do To Improve Conditions.

CAR IMPROVEMENTS CONSIDERED

Cleveland Section Hears About Future Automobiles and Braking Safety

Present and possible future improvements in automobiles were reviewed by Herbert Chase, of the Erickson Co., and the relation of four-wheel brakes to driving safety was dealt with by John R. Cautley, of the Bendix Brake Co., at the monthly meeting of the Cleveland Section which was held on Feb. 15 in the Hollenden Hotel. The meeting was preceded by a dinner, and the delivery of the papers was followed by questions and discussion by interested members, who were in attendance in gratifying number.

From the outside in was the method of treating his subject that was adopted by Mr. Chase, who began logically with the finish of the present new cars and followed through the apparently probable future developments of the various parts down to the rims of the wheels. The development of nitrocellulose lacquers has worked a revolution in finishing, he said, with a material saving in production cost and a great increase in durability of finishes. Many of the finest bodies in the recent shows, however, had varnish finishes or varnish top-coats over lacquer that gave a depth of luster that is not commercially obtainable as yet with lacquer top-coats. Reduced prices of closed cars indicate, said Mr. Chase, that body builders have adopted the rapid production methods that have long characterized chassis production. Little progress has been made, however in reducing body weight by the adoption of flexible bodies as abroad, which would permit of reduction of chassis weight. If bodies were relieved from chassis stresses by some form of flexible mounting on the frame and if a light, flexible substitute for plate glass were available, considerable saving in weight could be effected. All-metal bodies have certain advantages and if aluminum production were increased and the price sufficiently reduced, both the chassis and the body could be made much lighter.

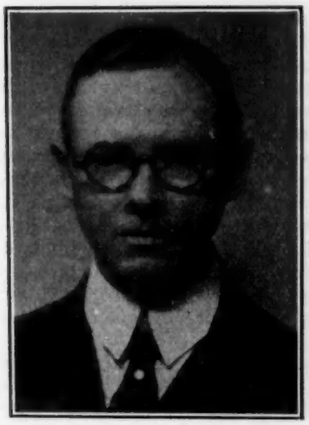
Some Developments That May Come

There is an evident tendency toward lower chassis and bodies, observed the speaker, and some attention has been given to adjustable driving-seats. Better heating and ventilating are needed, and windshields should be more easy to adjust. There is also room for improvement in interior finish and upholstery so that cars will be easier to keep clean inside. Four-wheel brakes, eight-in-line engines, balloon tires, and better chassis lubrication have won popular approval.

Possible future developments that may become common practice include lower-speed engines, two-cycle constant-compression engines, oil-filters, air-cleaners, steam or evaporative cooling, central chassis-lubrication, welded-tube chassis frames, a torque-multiplying device having continuous and infinitely variable power transmission, elimination of the clutch, self-locking differential, vacuum-operated brakes, pneumatic suspension, non-metallic spring-shackles, and drop-base wheel-rims.

Safety the Prime Requirement

Today is the day of brake study, asserted Mr. Cautley, who declared that safety is the paramount requirement of present automobiles and that, while acceleration and the control of steering have been given much attention for years, the control of deceleration is equally vital. He supported his



Herbert Chase

John R. Cautley

contention by quoting replies received from 28 engineers to a request for their opinions, as expert drivers, of the chief advantages of four-wheel brakes. Practically all either directly or indirectly emphasized the safety value of this type of brake. One man reported that in runs of 418 miles on hard roads over the Allegheny Mountains an average speed of 39.5 m.p.h. was maintained with four-wheel brakes and a maximum speed of 55 m.p.h as against an average speed of 37 m.p.h in the same car with two-wheel brakes and a maximum speed of 70 m.p.h.

With two-wheel brakes, said Mr. Cautley, a deceleration greater than 9 ft. per sec. per sec. is uncommon and stops at rates in excess of 15 ft. per sec. per sec. begin to be uncomfortable; nevertheless, stops with a deceleration rate of from 24 to 27 ft. per sec. per sec. are entirely possible. This tremendous deceleration is available as a "safety braking reserve." The passengers may be made uncomfortable but "they do not go to the hospital or cemetery." The higher safe speed made possible by four-wheel brakes, he said, means higher minimum speed on the highways and will be an important factor in reducing the increase of traffic congestion.

Discussion Covered Wide Range

Discussion following delivery of the two papers related mainly to steam-cooling but also touched upon four-wheel brakes; improvement of, or a substitute for, sliding-gear transmissions; use of superchargers on passenger cars; regulation of the charging rate from generators; and the need for giving attention to the more efficient use of fuel in the engine. Among those who participated in the discussion were the authors of the papers, Chairman John H. Jaschka, J. W. Saffold, Ernest Wooler, George W. Albright, P. D. Manbeck, Ferdinand Jehle, S. K. Wellman, T. V. Buckwalter, Glenn A. Toaz, Romuald Karasinski, and E. Gruenfeldt.

LATEST DATA ON STORAGE-BATTERIES

Modern Starting and Lighting Batteries Described to the Washington Section

Beginning with the early history of the storage-battery and tracing its development up to the present, Dr. G. W. Vinal, of the Bureau of Standards, devoted the major part of the subsequent text of the paper on Storage-Batteries which he presented at the Feb. 5 meeting of the Washington Section to details of the batteries now used for motor-vehicle starting and lighting. He showed numerous lantern slides illustrative of the variations in capacity, electromotive force and the like with temperature, of rates of charge and discharge, and kindred essential functions.

Regarding the rating of storage-batteries for automotive use, Dr. Vinal enumerated some of the reasons that justify the present ratings approved by the Society and also de-

scribed storage-battery manufacture. He emphasized the importance of making acid-tight the seal used where the terminal post passes through the cover of the battery cell, thus preventing battery-operation troubles that otherwise will occur later. He then explained exactly what takes place when a storage-battery is being charged and when it is being discharged. Comparative curves that resulted from data obtained by partly discharging a fully charged storage-battery of about 300 amp-hr. capacity and then charging it successively by the constant-current and by the constant-potential methods were exhibited. If the proper control of the charging energy is assured, whether it be current or voltage control, Dr. Vinal thinks no reason exists for judging that either method of charging is not as good as the other.

Commenting upon the respective capacities of thin-plate and of thick-plate batteries and the frequent report that the former type has higher capacity, Dr. Vinal pointed out that a greater number of thin plates can be put into a given space than if the plates are thick, which conveys the impression that the capacity must be greater, and said that as the discharge period becomes longer and longer, thick-plate batteries become more and more effective but that, at high rates of discharge such as 10 to 20 min., it makes little difference whether the plates are thin or thick. Concerning the desirable degree of concentration of the electrolyte, a cell was tried out successively using different degrees of concentration and it was found that, as the degree of concentration was reduced, the output of the battery was decreased very considerably, which emphasizes why the electrolyte in an automotive storage-battery that must deliver relatively heavy currents must be up to a reasonably high degree of concentration.

Among the other features of interest set forth by Dr. Vinal were lantern slides and accompanying descriptions bearing upon the following subjects: Rates of discharge, the effects of impurities in the electrolyte, sulphation of battery plates, specifications for acid and for electrolyte, freezing of the battery solution, and various temperature effects. A group of views illustrating the work a storage-battery has to do on an automobile and what happens when the starting-motor pedal is depressed was shown also.

Points of special interest brought out during the discussion of the paper had reference to the methods of charging, to the porosity of different types of separator, to the effects of thick and of thin plates upon the length of life of the battery and to the different requirements of different types of battery service. The majority of the members and guests present participated in the discussion.

NORTHERN CALIFORNIA SECTION ACTIVE

Automotive Topics Discussed at Weekly Luncheons as Well as at Monthly Meetings

An ambitious program is laid out each month by the Northern California Section. In addition to a regular monthly meeting, a weekly luncheon is held every Tuesday at noon, at which time some topic of interest to automotive engineers is discussed. Much credit is due to the officers who are responsible for planning and conducting the meetings and to the members whose enthusiastic support makes these enjoyable gatherings possible.

The Wizardry of Radio

The monthly meeting of the Northern California Section was held at the Engineers Club, San Francisco, on Feb. 18 at 6:30. The technical session was preceded by a dinner. Mr. Thompson, of the General Electric Co., who was in charge of the meeting, showed three motion pictures which were greatly enjoyed by those present. The motion pictures were entitled, respectively, the Wizardry of Radio, Beyond the Microscope and Intimate Associates of Thomas Edison.

A Talk on Spark Plugs

The weekly luncheon on Feb. 2 was held as usual at the Engineers Club, 47 members and guests attending. Chairman Wood, presiding, introduced Mr. Jarman, of the Champion Spark Plug Co., who gave some interesting facts relative to the requirements and the functions of the spark-plug in relation to the efficiency of internal-combustion engines. Mr. Jarman outlined the qualities needed in an insulator and told of the development of a material for this purpose which is used in all the plugs produced by his company. This material, sillimanite, has been made synthetically on a commercial scale, but his company is now using the natural form from a large deposit in California, which deposit so far as is known is the only one in the world of the required purity. The speaker stated that spark-plugs should be changed when the deposit of metallic oxides on the insulator tends to cause a leakage path for a portion of the ignition current. While such deposits would vary under different conditions, it would probably be wise, he said, to put in new plugs at least once a year or at the expiration of 10,000 or 12,000 miles.

How To Improve Lubrication

At the next luncheon, which was held at the Engineers Club on Feb. 9, Chairman Wood welcomed 18 members and guests and introduced W. W. MacDonald, of the W. W. MacDonald Co., who gave an interesting talk on the subject of oil rectifiers and filters. Mr. MacDonald believed that, in an endeavor to obtain reasonably perfect lubrication, the dilution phase of the problem is of far more importance than the filtering. He advocated rectification rather than filtration as a means for purifying the oil, stating that 40 per cent of the sludge filtered out of the oil is water and that this water, as well as fuel, can be more easily removed by rectification than by filtration. In the use of a rectifier, a simple cotton filter may be interposed, he said, to remove the solid contamination. The speaker called attention to the impracticability of attempting to rectify the entire oil stream of an engine and said that the company with which he is connected is satisfied to maintain the dilution equilibrium at about 10 per cent.

OIL RECTIFIER USE ADVOCATED

Skinner, Wall and Coulombe Describe Devices and Explain Results to Indiana Section

According to the three principal speakers at the Feb. 11 meeting of the Indiana Section, an automobile engine that is unequipped with a suitable rectifying device is no place for a "self-respecting oil that wishes to remain pure." This meeting, at which various types of contamination and methods of purifying were described, was advertised as a three-ring circus, and it developed so as to justify its name. In accordance with the long established practice, the meeting started on time and finished on time under the able guidance of Chairman George T. Briggs.

Skinner Explains Rectification Results

After reviewing briefly the principal items of crankcase-oil contamination, Ralph. L. Skinner, of the Skinner Automotive Device Co., Inc., mentioned the deleterious effects that result from the presence of fuel, water, acid, and solid material. He called attention to the fact, however, that when the fuel end-points are removed and the water is eliminated, the crankcase oil maintains sufficient body to avoid excessive wear by the solid matter that consists of metal particles, carbon and road dust. With the water eliminated, there is little opportunity for sludging, and the formation of acids that etch the metal parts is avoided.

Mr. Skinner presented, by slides, conclusive evidence that contamination can be so handled by a suitable device as to minimize any ill effects that would otherwise accrue. Briefly, the device by which these results are accomplished consists of a rectifier that receives the diluted oil from behind the lower ring and subjects this oil to a properly regulated heat-treatment that drives off the fuel heavy ends and the water and returns the rectified oil to the crankcase. The action of the device is intermittent and gains its energy for operation from the suction of the intake-manifold.

The importance of a correct application of heat to the oil was emphasized, in view of the necessity for removing the contaminants without carbonizing the oil itself. This control of heat is brought about by the use of a thermostatic device.

After presenting a large number of analyses of used oils that had been rectified successfully, and after demonstrating by means of results accomplished the fact that without dilution undue wear does not occur, Mr. Skinner concluded with the recommendation that all new internal-combustion engines be provided with some means for the positive prevention of dilution; that an oil of correct viscosity, pour test and flash point be used, and that all means be directed toward the conservation of petroleum,—an irreplaceable natural resource.

Wall Describes Combination Filter and Rectifier

Following Mr. Skinner's paper, William Guy Wall, creator of the Three-Way Oil Purifier, described his device that provides for the removal of dirt by filtration and also the removal of other contaminants from the crankcase oil by a suitable application of exhaust heat. The oil is kept in circulation by a pump, the rate of circulation being about 3 gal. per hr. An asbestos cloth filter is employed to remove the particles of dirt.

In discussing the removal of the fuel heavy ends from the contaminated oil, Mr. Wall called attention to the fact that the time element is important in the determination of the degree of heat necessary to volatilize the diluent. He also stated that the diluent becomes volatilized more readily under the reduced pressure that exists in the rectifier.

An interesting feature of the device described by Mr. Wall is the siphon, by which the filtered and rectified oil is returned to the crankcase.

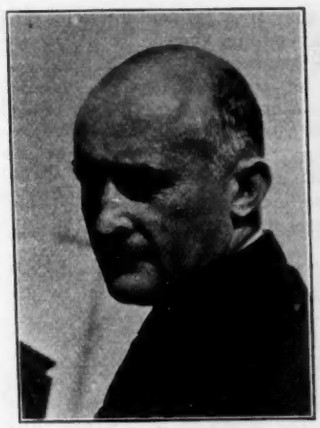
Retifier Utilizing Vacuum and Heat Described by Coulombe

Joseph C. Coulombe, of Byrne, Kingston & Co., described an oil rectifying device that employs a heat unit which is strapped to the exhaust-manifold and a vacuum tank arrangement that is actuated by the vacuum from the intake-manifold which brings about the satisfactory flow of oil through the heat element. The apparatus is intermittent in operation, like the conventional vacuum tank, and the heat element is always submerged. The vacuum tank is provided with a float-actuated dumping device that allows the oil to pass from the upper to the lower chamber. Aeration of the oil is provided for through both the filling and the dumping cycle. The lower chamber of the vacuum tank contains a filter through which the rectified oil passes on its way back to the crankcase. An overflow tube is provided in this chamber to insure satisfactory operation even when the oil is too viscous to pass through the filter.

In discussing the matter of heat application and control Mr. Coulombe showed how the temperature may be controlled during winter and summer by mounting the heater block on a shunt of the exhaust system. By using valves the flow of exhaust gas through this shunt can be regulated from the dash.

George T. Briggs

William G. Wall



Ralph L. Skinner

Joseph C. Coulombe

Among those who participated in a very spirited discussion of the three interesting papers were Messrs. Duesenberg, Halloway, Teetor, Crawford, Geertz, Berry and Staley.

INTERESTING ENGINE COOLING MEETING

Southern California Section Hears Two Papers and Enjoys Lively Discussion

The cooling of automotive engines was the subject interestingly handled at the meeting of the Southern California Section on Feb. 12, at the City Club, Los Angeles. The meeting was preceded by a dinner which was attended by 85 persons. After the dinner and a musical program, the meeting was called to order by Secretary Ethelbert Favary, who presided in the absence of the Chairman and the Vice-Chairman of the Section.

Technical Questions Should Be Studied

G. A. Dockeray, of the Eagle Radiator Co., presented the first paper on the program. Stating in his introductory remarks that less attention has been devoted to engine cooling than to any other feature connected with the design and construction of automobiles, trucks and airplanes, the speaker emphasized the need for more efficient cooling. He pointed out that California presents varied and extreme conditions which add to the difficulty of the problem, with a consequent need for a thorough study of the technical questions involved in order that adequate cooling may be obtained.

Types of Cooling System Discussed

Paul H. Zehring, of the Cartridge Mechanical Works, agreed with Mr. Dockeray that relatively little advancement has been made in the development of cooling systems in comparison with the improvements in the other parts of automotive vehicles. He pointed out, however, the possibility that the great strides that have been made toward better combustion, better suspension, the lessening of friction and wear, and the elimination of weight in reciprocating parts have made the need for radiator improvements somewhat less imperative.

The speaker named seven types of engine-cooling system, commenting briefly upon each. He mentioned several causes of boiling in the cooling system and, after stressing the need for keeping the radiator clean, he described a method for washing out radiators on automobiles and trucks.

Many Participate in Discussion

Opening the discussion, Mr. Favary spoke of the problem as a complicated one, stating that formerly a cooling

system was considered satisfactory if it prevented the engine from overheating, but today a cooling system is called inefficient if it keeps the engine too cool.

Steam-cooling was a phase of the subject that received considerable attention. Laboratory tests were described in which steam-cooled engines running over full load delivered in excess of 30 per cent thermal efficiency whereas the same engine if operated with an ordinary radiator would have delivered from 22 to 24 per cent. A further advantage to be derived from steam-cooling, it was stated, was the fact that the temperatures remain uniform.

Some interest was expressed in the cleaning of radiators. The use of steam as a cleaning agent was advocated by one person who said that steam had the necessary force to do the job, without producing injurious effects. A suggestion was made that the radiator could be efficiently flushed by replacing the lower hose by a hose from a faucet or water line, taking the radiator cap off and forcing the water up from the bottom. The use of acids in cleaning radiators was advocated by several persons, who stated that alkali must be used to remove oil or grease but that an acid must be used to counteract whatever alkali may remain in the system.

Relative to the depth of radiator core, Mr. Zehring told of a test which indicated that efficiency increased with additional depth of core.

Mr. Favary announced that the next meeting of the Southern California Section, scheduled for March 12, would be devoted to a consideration of brakes, brake construction, brake up-keep, brake adjustment, and methods of computing how quickly an automobile should stop and what determines the stopping.

TRAFFIC CONGESTION AND ITS RELIEF

Small Cars, Motorcoaches and Other Remedies Discussed at Milwaukee Meeting

A. W. S. HERRINGTON

Some of the problems arising from traffic congestion in large cities were presented by A. W. S. Herrington, recently chief engineeer of the Motor Transport Division of the Quartermaster Corps but now an automotive consulting engineer, at the meeting of the Milwaukee Section that was held on Feb. 3. The address was illustrated with about two dozen lantern slides showing traffic conditions in Chicago, flow of traffic into and out of New York City, and the relative efficiency of street-cars, motorcoaches and private passenger-cars.

Without venturing to suggest a cure-all for the traffic troubles, Mr. Herrington emphasized the need of segregation of the different classes of traffic, of placing intersecting traffic on different levels and of standardizing traffic regulations in all cities. Following the address, there was much animated discussion by attending members, in which the small car, the motorcoach, motor trucking of short-haul freight, and traffic regulation by light-signals were noticeably in mind as possibilities for alleviating conditions.

City Traffic Studies by Engineers Needed

It is vitally necessary that at the present time a study of the future requirements of all forms of transportation should be under way in every community, said Mr. Herrington, and this should be in the hands of competent engineering authorities. Cities in which this is done will be much farther ahead 10 or 15 years hence than those that ignore the problem now and that later will find themselves in such a condition that they will have to go through growing pains which will not be agreeable.

Statisticians have tried to determine the automobile saturation-point in terms of the number of persons possessing money enough to afford a car, but it can be definitely found, he said, in a simple arithmetical determination of the number of square feet of road space that we have on which to operate vehicles. Manhattan Island was cited as an example in proof of the theory. Bounded by water on all sides, its population has decreased from 2,800,000 in 1910 to 2,500,000 in 1920 and now remains stationary. The trend of motor-vehicle registration continues upward but the average for the borough is 100 vehicles per 1000 persons, whereas for the City of Greater New York it is 260 per 1000.

The time factor is intimately related to space limitation, said Mr. Herrington. When a single vehicle is involved it is a simple matter to determine its average rate of speed by a time-distance curve, but when two or more are proceeding in the same direction at the same time and frequent stops and starts are necessary, it is not so simple. The rate of acceleration of the average car which is 15 ft. long is 6 ft. per sec. per sec. and the deceleration rate is 8 ft. per sec. per sec. Based on these factors, a single line of unmixed and uninterrupted traffic reaches its maximum discharge at approximately 15 m.p.h. This theoretical rate never is realized in practice. Fifth Avenue, with its block system of light-signals, allowing 90 sec. for traffic movement on the avenue alternating with 60-sec. intervals for flow of cross-traffic, has a theoretical capacity of 975 vehicles per lane per hour but, due to mixed traffic and other conditions, this is reduced actually to 750 per hour, or 77 per cent of the theoretical capacity.

Street-Cars Must Be Eliminated

Grade separation may prove to be the only alternative in some of the centers of heaviest traffic, such as New York City, as the most serious congestion arises from the cross-flow and the intermingling of different classes of traffic. Street-cars operating on rails in the center of the street must be eliminated, said Mr. Herrington, who cited the case of the Detroit & Port Huron Shore Line Railway as an example of forces that will bring this about. The cars formerly ran to the center of Detroit but, when a concrete road was built parallel with the railway, it became difficult to compete with motorcoaches, operated on the highway, which could make better time. A study of the situation showed that the electric-cars lost most time inside the city limits, so the traction company built a new terminal at the limits and put on motorcoaches to run to the center of the city. By this combination, the electric line could make better time to Port Huron than the competing motorcoaches and recovered its patronage.

The double-deck motorcoach is the most efficient user of the street and carries about 2 1/3 times as many persons per square foot of street space occupied as the 59-passenger street-car and nearly 5½ times as many as the private automobile. The taxicab also is a much more efficient vehicle than the private car, as more than 50 per cent of its mileage is passenger-carrying mileage.

If traffic increase continues at the present rate, it will become an almost unsolvable problem by 1965, as shown by a chart, as the increase in the various traffic lanes will be from 450 to 960 per cent over present lane discharges.

Nomination for Section Officers

At the conclusion of the prolonged discussion, the meeting elected Fred M. Young to represent the Milwaukee Section on the Nominating Committee of the Society for nomination of officers at the Summer Meeting at French Lick Springs in June. Upon request of Vice-Chairman George W. Smith, who presided, Secretary A. C. Wollensak announced the nominees for officers of the Section, as follows: Chairman, Walter Ethan; Vice-Chairman, Joseph P. Armitage; Secretary, Arthur C. Wollensak; and Treasurer, George C. Apple.

The next meeting of the Section is to be held March 3,

when F. Jehle, research engineer of the White Motor Co., will present a paper on Laboratory Methods and Technique.

FACTORY SHOULD CONTROL SERVICE

Hutt Talks on Service-Station Work to Service Managers at Philadelphia

The day is approaching rapidly when all automobile builders will realize that they must exercise some direct control over the service-stations for their cars, asserted A. E. Hutt in a paper delivered by him at the Feb. 9 meeting of the Pennsylvania Section at Kugler's Restaurant in Philadelphia following a dinner of attending members and their invited guests from the Automotive Service Association, most of whom are service managers. The guests entered freely into the discussion after the presentation of the paper. Chairman C. O. Guernsey presided.

A. E. Hutt

Mr. Hutt also said that he believes that car agencies will, in time, be granted by builders only under contracts that permit the factory to dictate the methods to be followed in the maintenance of their vehicles and the tool equipment required for the shop and the mechanics and to insist upon receiving reports on all jobs handled, written on forms provided for the purpose. Two distinct advantages to be gained by the factory from the control of properly organized service-stations are (*a*) maintenance of customers' vehicles in first-class condition at the minimum cost to the owners, which assures satisfaction and repeat orders and (*b*) ability of the factory to obtain and tabulate reports on mechanical failures and needed adjustments on the cars it produces. These reports will provide a valuable reference file for the engineering department, especially when considering changes to be incorporated in new models.

What the Factory Should Do

Foremen of the mechanical department of the service-stations should spend some time in the factory and be approved before being given supervision of mechanics. Instructions should be furnished by the factory for every conceivable job to be done in the service-station, and the tools to be used on each job should be specified, since many so-called mechanics have had little training and hardly know how to use any but the most ordinary tools. Every mechanic should be obliged to obtain the necessary hand tools for the work he is to do. These should preferably be supplied by the service-station but paid for by the mechanic (*a*) because no man should be permitted to work without the necessary tools and (b) to assure uniformity and quality of the tool equipment. The ordinary tools could be supplied at cost and any necessary expensive tools could be drawn from the tool-room on a check system.

The need for providing an instruction book for doing jobs, said Mr. Hutt, is due to the fact that no two mechanics proceed to do a job in the same way and too often they take-down a car more than is necessary, which causes needless expense to the owner. There may be various ways of getting at any part of a car to make adjustments, replacement or repairs, but there is only one easiest and best way, and that way should be prescribed by the factory and followed by the service-station. Thus the mechanics will become adept in it, fixed prices can be charged for the job and the station can check-up on the time the mechanics spend on jobs of that particular kind.

Could Insist on Quality Work

By exercising control the factory can insist upon cleanliness, requiring that grease and dirt be removed from the car before work is started on it and that mechanics shall not smear grease, oil and dirt upon the body, cushions, steering-wheel and controls; that they shall not remove tools and cloths from the car to use; that all jobs shall be done in the most approved way and in accordance with

SCHEDULE OF SECTIONS MEETINGS

MARCH

3—Milwaukee Section—Laboratory Methods and Technique—Ferdinand Jehle

5—Washington Section—The Public Mind and the Modern Motor Car—C. H. Warrington

9—Pennsylvania Section—Fleet Management—Col. R. O. Mason; also T. O. Day

10—Dayton Section—Spring-Suspension and Riding-Qualities—F. C. Mock

11—Detroit Section—Engine Design and Anti-Knock Fuels—S. W. Sparrow
Indiana Section—Motorcoach Transportation—A. J. Scaife

12—Southern California Section—Brakes and Brake Developments—John Wiggers; Mathematical Analysis of Brake Requirements—E. Favary

15—Cleveland Section—Research—Charles F. Kettering

16—Buffalo Section—Studies on the Oscillation of Leaf Springs—N. E. Hendrickson

18—Metropolitan Section—Fuels and Lubricants: The Fuel Situation from the Producer's Viewpoint—A. J. Cramer; Present and Possible Improvements by the Automotive Manufacturer—S. W. Sparrow; What the Operator Can Do to Improve Conditions—J. Willard Lord

25—Detroit Section—Automobile Gearing—John Bethune

26—Chicago Section—Development and Trend in Industry—F. E. Moskovics; Progress in Engine and Chassis Design—H. L. Horning; A Suggested Remedy for Crankcase-Oil Dilution—R. E. Wilson
Northern California Section—Gasoline and New Fuel Tests—William S. James

APRIL

7—Milwaukee Section—Chassis Lubrication—Fred Gleason

8—Detroit Section—C. F. Kettering
Indiana Section

19—Cleveland Section

22—Detroit Section

factory instructions, and that coach work performed outside of the service-station shall be of good quality.

The factory is in a position to know what special tools, jigs and set-ups are required to maintain its vehicles and should require all of its service-stations to obtain them. It should instruct the foremen in their use and specify the wrenches and other small tools needed for the different units of the vehicle. Blueprints of all units, sub-assemblies and the general assembly should be provided, together with a large chart of the lubrication points and the types of lubricant to be used. There should also be a large phantom chart of the wiring system and printed instructions with numbered or lettered references to the blueprints and charts.

AIRPLANE-ENGINE MAINTENANCE

Mechanical and Personal Causes of Forced Landings Analyzed for Detroit Section

LIEUT. C. W. BETTIS, U. S. A.

An airplane engine depends upon four distinct and separate systems mounted on the outside of its engine for its successful operation during flight; namely, the ignition, the water-cooling, the fuel, and the oiling systems. Since the failure of any one of these systems usually necessitates a forced landing, they should be given the maximum amount of attention while the airplane is on the ground awaiting its next flight, according to Lieut. Cyrus Bettis, winner of the Pulitzer Race and famed pilot of Selfridge Field, now engineer officer of the 95th Pursuit Squadron of the Army Air Service, who delivered his paper on Airplane-Engine Maintenance at the Detroit Section meeting that was held on Feb. 25. A dinner preceded the well-attended meeting, of which W. R. Strickland was chairman, and the speaker was introduced by L. M. Woolson. As Lieutenant Bettis traced the causes, both personal and mechanical, of forced landings and related personal experiences, he aroused and held continuously the interest of the assembled members and guests.

Especial need for design and construction that will eliminate mechanical and electrical sources of possible trouble was expressed by Lieutenant Bettis, who stressed also the necessity of frequent and thorough inspection of all airplane parts. That we are making progress in aviation in this Country is indicated by the 1925 report of the Surgeon General of the Army, in which it is stated that the number of crashes per flying hours of heavier-than-air machines decreased from 1 crash per 200 flying hr. in 1921 to 1 crash per 500 flying hr. in 1924; and that there was a decrease from 1 fatality per 955 flying hr. in 1921 to 1 fatality per 3365 flying hr. in 1924. The reduction can be traced to the existence of a flying personnel having greater experience and to improved airplanes and engines.

It was noted further by the speaker that out of the 256 crashes during 1924, there were 58 charged to engine failure and 29 due to faulty maintenance, or approximately 33 1/3 per cent. Poor piloting and inexperience constitute the one other major cause of crashes, being responsible for another 33 1/3 per cent, but it is believed that this percentage will be reduced as pilots become better trained and gain greater experience. Crashes due to engine trouble can, however, be reduced in number only by developing better engines and devoting greater effort to their installation and maintenance. If the two main causes that account for 66 2/3 per cent of the crashes can be eliminated and all other causes remain as they are now, a reduction to something like 1 fatality per 10,000 flying hr. could be attained which, allowing an average flight-distance of 100 m.p.h., would be equivalent to but 1 fatality per 1,000,000 miles of flight. In fact, an airplane engine seldom fails of itself, in flight; the failure usually is caused by neglect or improper attention prior to flight, which led Lieutenant Bettis to treat in specific detail the proper methods of maintaining the four vital systems which are exterior to the engine and upon which its reliability depends.

Prepared notes that had been submitted by Arthur Nutt, chief motor engineer of the Curtiss Aeroplane & Motor Corporation, were read, in which he stated his belief that the present high percentage of failures is due to the use of old equipment. With modern machines, he said, much better records are possible. He recommends the use of water coolers for the engine oil in preference to air coolers, and advocates also the adoption of recommended practices in airplane design and construction features rather than the use of definite standards that tend to kill progress and stifle originality. Lieutenant Bettis remarked that the Army is rapidly doing away with the old-type wooden construction for airplanes and is substituting modern metal construction.

One of the most interested listeners in the audience was 13-year-old Frank Rippingille, son of E. V. Rippingille. The boy already has some 15 hr. of flying time to his credit, a goodly percentage of which is solo flying.

AIRCRAFT-ENGINE PROGRESS

Ten-Years' Development of Curtiss Engines Reviewed at Buffalo Meeting

ARTHUR NUTT

The latest trend in aircraft-engine design and present and probable future performances of airplanes in speed, ceiling and endurance were discussed most entertainingly by Arthur Nutt, chief motor engineer of the Curtiss Aeroplane & Motor Corporation, at the Feb. 23 meeting of the Buffalo Section at the Statler Hotel in Buffalo. About 80 members and guests were in attendance and there was considerable discussion after the paper was presented.

After reviewing in detail the progress in design of Curtiss water-cooled engines since 1916, which has resulted in the production of an engine that develops 1 hp. per 1 10 to 1.33 lb. of weight and is asserted to be 50 lb. lighter than its nearest competitor either here or abroad, Mr. Nutt stated that the company last year undertook the production of a 400-hp. nine-cylinder radial air-cooled engine. This, as well as the water-cooled types, was shown in lantern slides. It is a conventional type of radial engine and the present development progress indicates that it will be a very useful type. It develops 400 hp. at 1650 r.p.m. and weighs 800 lb., dry, or 2 lb. per b.hp., which is more than the dry weight of the 500-hp. Curtiss V-1400 water-cooled engine, namely, 685 lb. The installed weight of the two engines is about the same, but the weight of the radial engine can be reduced to 700 lb. as the result of experience gained in the last 2 years. The cylinder of the air-cooled engine was developed by the engineering division of the Air Service at McCook Field.

As the power of engines is increased above 500 hp., the weight per horsepower of air-cooled and water-cooled engines, installed, will be about the same, but above 1000 hp. the water-cooled engine will begin to have the advantage in installed weight.

Forty Different Materials Used

The problem in designing aircraft engines is to secure the best balance between weight, frontal area, cost, bulk, high performance, and reliability, said Mr. Nutt, and all of these problems are subject to the proper arrangement of parts and the proper selection of materials. More than 40 different materials are used in the modern aircraft engine, the most interesting of which are the light alloys. Magnesium alloy is one of the most unusual. It is 40 per cent lighter than aluminum, has an average tensile-strength of 20,000 lb. per sq. in. and an elongation in 2 in. of 4 per cent. It is comparable with the best non-heat-treated commercial aluminum crankcase alloys. This alloy has been cast in almost every form that is used in an aircraft engine but present cost of production prohibits its use in quantity.

Aluminum bronze has come into extensive use in aircraft engines and is a very remarkable valve-insert metal, as its coefficient of expansion is about the same as that of aluminum. It can be heat-treated to give about 200 Brinell hardness. Duralumin is also used extensively in present engines, both in cast and forged form, and is particularly advantageous in the latter form because of the small amount of work necessary on the forgings after they are received. Cast high-tensile aluminum alloy has been found unusually satisfactory for cylinder-heads, as it has an average tensile-strength of 30,000 lb. per sq. in. and an elongation in 2 in. of 6 per cent. It is used for cylinder-heads and water-jackets on the water-cooled engines and for crankcases on the air-cooled engines. It is approximately 10 per cent lighter and nearly 100 per cent stronger than the regular crankcase aluminum alloys. A bearing material having remarkable properties is the Y-alloy of aluminum with copper, nickel and magnesium. This is also used for pistons and in cylinder-heads in the air-cooled engines.

The Probable Future Trend

Future development of aircraft engines probably will be toward the combination of reduction gearing from crankshaft to propeller, high engine-speed and supercharging, according to Mr. Nutt. The reduction-gearing problem has been solved and the air-cooled engine is already equipped with a supercharger. This combination, with the use of light alloys, will make a weight of 1 lb. per hp. well within the possibilities. At present the compression-ratio is limited by commercial aviation fuel to 5.5 to 1.0. No radical change in power of the normal engine is to be expected except by raising the engine speed or by the use of a better fuel. It is expected that by the use of superchargers the mean effective pressure can be kept around 140 lb. per sq. in. at the higher speeds, which will mean a tremendous power output from a small engine.

SLEEVE-VALVE ENGINE DEVELOPMENTS

Dayton Section Hears Interesting Paper by P. M. Heldt, of *Automotive Industries*

"Of the many types of sleeve-valve engine on which patents have been issued, only two have come into extensive use,—the double-sleeve type, invented by Charles Y. Knight, and the single-sleeve type, invented by Peter Burt." Continuing with an historical summary thus introduced, P. M. Heldt, engineering editor of *Automotive Industries*, presented a very complete and interesting address on Sleeve-Valve Engines before the Dayton Section at its meeting on Feb. 10.

Reasons for Sleeves

The primary object of Mr. Knight, who pioneered the double-sleeve developments, was to render the engine more quiet in operation by making the valve action positive. Included among other advantages claimed are the favorable form of combustion-chamber, comparative freedom from carbon troubles and the absence of the often troublesome exhaust-valve head.

Many Interesting Points Covered

It is impossible in the space available for this news account to discuss even superficially the many interesting points that were brought out by Mr. Heldt under the following headings: Valve Action, Combustion-Chamber Shape, Permissible Compression, Riedler's Endurance Tests of Knight Engines, Port Opening Area, Timing of Sleeve Valves, Steel Sleeves, Junk Rings, Balance of Sleeves, Power Required by Sleeves, Detonation Characteristics, Lubrication, Sleeve Clearances, Circulation Through Head Jackets, Mercedes-Knight Engines, Single-Sleeve Engines, Form of Ports, Inlet Ports, Double Port in Sleeve, and the Vauxhall Single-Sleeve Engine. The paper is printed in full in this issue of The Journal, beginning on p. 303.

EVALUATION OF FUEL-KNOCK INTENSITY

Recent Audibility Anti-Knock Tests of Fuels Described to the Chicago Section

Prof. Daniel Roesch

All fuels have specific "knocking" properties which affect the limits of compression and consequently the power and the economy of the internal-combustion engine and, since the knowledge of these properties should disclose the fuels that are suitable for high compressions and result ultimately in improved engine-design and better fuel-economy, the object of the paper delivered by Daniel Roesch, associate professor of gas engineering, Armour Institute of Technology, at the meeting of the Chicago Section that was held on Feb. 19, was to describe the method adopted in the automotive laboratory of the Institute for ascertaining the knocking properties of various fuels and to set forth the results obtained from an extended series of tests. The fuels were used in a four-cylinder poppet-valve engine of the automobile type, having an L-head combustion-chamber and a compression-ratio of 4.5 and its carbureter having means for regulating the light-load and the full-load mixtures. A fixed carbureter-adjustment was used mainly. Connection between the engine and a cradle dynamometer was through a double universal-joint and the customary methods of determining speed, torque and temperatures of the air, water and oil were employed.

In summarizing the test results, Professor Roesch said he believes that the methods employed furnish valuable data pertaining to the anti-knock qualities of fuels, combustion-chamber shapes and the efficacy of gasoline dopes, and that

(*Concluded on p.* 251)

AUTOMOTIVE RESEARCH

The Society's activities as well as research matters of general interest are presented in this section

RIDING-QUALITIES

Subject for Cooperative Research. Instruments for Establishing Fundamentals

In negotiating the sale of automobiles the prospective customer is usually given a ride so that he may judge the riding-quality and flexibility of the vehicle under consideration. In view of this fact, it is remarkable that so far only few manufacturers have made an attempt to develop instruments for the measurement of the riding-qualities of their product. Graphic records are made of engine performance, acceleration and deceleration, hill-climbing ability, and maximum and minimum speeds of the car; but on riding-qualities the opinion of one or more executives is considered as final. If a dispute arises, so far no reliable measurements could be carried out by the engineering department to serve as predominating factors in a final decision.

Inasmuch as springs and tires have a profound influence on the riding-qualities of a vehicle, it is very natural that all manufacturers should be careful to specify the desired characteristics of the springs and the size and the inflation-pressure of the tires. In so doing they are usually guided by a general desire for greater flexibility in both springs and tires. However, the advantages that accrue are by no means obtained without accompaning disturbing factors. This contention is well borne out by the numerous discussions pertaining to the various means conceived to dampen disagreeable motions of motor cars.

The riding-qualities of a vehicle, though largely determined by the springs, tires and damping devices, also depend to a large extent on the location of the center of gravity, the wheelbase, the ratio of sprung to unsprung weight, and the road contour. Because of the large number of factors that have to be considered, the problem can only be properly investigated and solved through cooperative research. Manufacturers of cars, tires, shock-absorbers, and springs, and officials from the Bureau of Public Roads and from the Highway Research Board, should participate in this work. It is obvious that, to assure progress in the cooperation of such a group, it is necessary first to establish fundamentals on which all should readily agree.

The Society is endeavoring to initiate adequate steps that will lead to cooperative research. It is hoped that in time the number of participants will increase. At present a noteworthy start has been made by F. C. Mock[1], of the Stromberg Motor Devices Co., and R. W. Brown[2], of the Firestone Tire & Rubber Co. In their work they use instruments that are suitable for recording accelerations of relatively high frequency.

There can be no doubt that the use of accelerometers for the measurement of riding-qualities is as important as, or more important than, the use of seismographs. However, it has been difficult to devise an instrument that will record variable accelerations as readily as a seismograph records variable displacements. Considerable thought has been given to this subject by Dr. H. C. Dickinson and Dr. Benjamin Liebowitz, who have been good enough to furnish the following description of two interesting instruments they have developed recently.

[1] See THE JOURNAL, July, 1925, p. 37.

[2] See THE JOURNAL, July, 1925, p. 107; and December, 1925, p. 546.

[3] See *Journal of the Franklin Institute*, August, 1919, p. 237.

[4] See THE JOURNAL, December, 1924, p. 558.

[5] See THE JOURNAL, December, 1925, p. 547.

RECENT DEVELOPMENTS IN CONTACT ACCELEROMETERS

The term "contact accelerometer" used herein has been suggested to designate the type which depends upon the *initial motion* of a weight pressed against a stop with a predetermined force exerted by a spring. Any appropriate elastic medium is included in the term "spring."

The fundamental principle of contact accelerometers may be summarized thus: if a weight at rest, of mass m, is pressed against a stop with force F, and if the system is subjected to an acceleration, a, in the direction of F, the weight and stop will begin to separate as soon as the accelerating force impressed on the weight exceeds the force F, that is, when $ma > F$ or $a > F/m$. Note that F here indicates the initial force between the weight and the stop; for a vertical system, F is the force exerted by the elastic support less the weight of the mass.

This principle is probably quite old, but has not received much engineering attention until recently. Zahm[3] appears to have been the first to use a *battery* of contact accelerometers, arranged so as to draw in effect a discontinuous "curve" of acceleration.

Considerable development work has been done in the last few years on accelerometers that depend upon the amplitude of motion of a spring-supported weight; but the indications of such instruments are not independent[4] of the periods of the impressed forces, and, to obtain reliable results, the period of the instrument must be made inconveniently small, or else separate determinations of impact periods must be made.

The basic advantages of the contact principle are

(1) Since only the *initial* motion of the weight is involved, contact accelerometers are, or can be made to be, independent of the periods of impressed forces, over a very wide range

(2) The method consists virtually in balancing a static reaction against a dynamic force, and is therefore susceptible of *static* calibration, and involves no uncertain quantities

It is thought that, of all the means available at present for the measurement of acceleration, the contract principle offers the simplest, most direct, and most reliable method.

METHODS OF INDICATION

The indications of contact accelerometers depend upon recording or otherwise observing the separation of weight and stop. This may be done in several ways. For instance, the motion of the weight may, by direct mechanical connection, move a stylus into, or preferably out of, contact with a moving strip of paper, as was done by Zahm.[3] But, as will be discussed below, the best method is believed to be the opening of an electric contact by the initial motion of the weight, as in Brown's accelerometer.[5] The opening of the contact may in turn be made to operate some form of indicating instrument, or counter, or other electro-mechanical recorder. In the tests described herein, the contact opening was determined, in most instances, by a telephone receiver.

BATTERY OF CELLS VERSUS SINGLE ADJUSTABLE CELL

For a given setting of its spring, a single contact-accelerometer cell indicates only when the corresponding value of acceleration has been exceeded. For some purposes, this is sufficient, as when it is desired to record the number of times a given acceleration is exceeded, under a given set of conditions.[5]

For general purposes, especially road work, when determinations of acceleration maxima are desired, a battery of such cells is required, each set, in properly graded steps, for a definite value of acceleration. Such a battery or group of cells, associated with proper recording means, offers a complete solution to the problem of direct measurement of acceleration, in all its phases, at least insofar as automotive applications are concerned.

On the other hand, for certain purposes, as in the laboratory where a given acceleration can be repeated a number of times with fair accuracy, or on the road, where the total number of accelerations exceeding a given value is sought, a single-cell *adjustable* contact-accelerometer will meet the requirements simply and inexpensively. This paper is intended mainly as a report on the design and testing of two such contact accelerometers. The principal points sought in both designs are

(1) Reliability
(2) Accuracy
(3) Ruggedness
(4) Wide range and "open scale"
(5) Simplicity and ease of fabrication

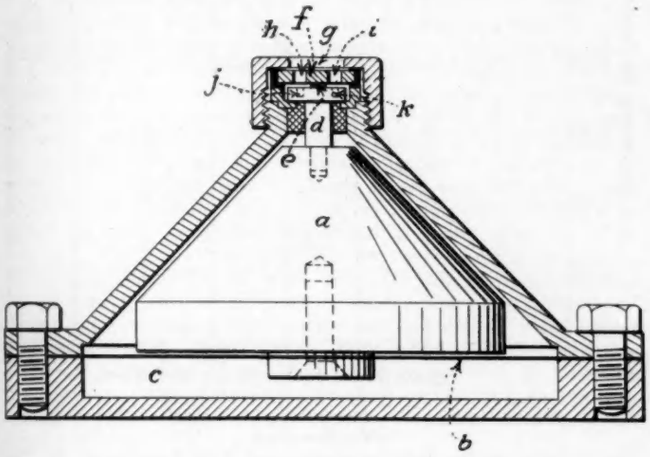


FIG. 1—CROSS-SECTION OF A DIAPHRAGM TYPE OF CONTACT ACCELEROMETER

The Inertia Element *a* of This Instrument Is a Truncated Conical Steel Weight Weighing 2¾ Lb. and Having a Total Movement of 0.001 In. The Complete Instrument Weighs Approximately 10½ Lb.

The diaphragm type of contact accelerometer was designed with the additional object in mind of securing maximum ease of setting and adjusting. In designing the micrometer type, one particular object was lightness and compactness, with a view to ultimate adaptation to battery construction.

DIAPHRAGM TYPE OF CONTACT ACCELEROMETER

Fig. 1 is a drawing showing a cross-section of the diaphragm type of contact accelerometer. It consists of a truncated conical steel weight, *a*, supported on a thin brass diaphragm of about 5 in. diameter, *b*. Provision is made for admitting air under pressure to the space *c* between the base and the under-side of the diaphragm. The outside diameter is about 6½ in., and the weight of the instrument without accessories about 10½ lb. The inertia element, including attachments, weighs about 2¾ lb.

The top of the weight *a* carries a steel stud, *d*, that has a sliding fit in a fiber sleeve carried in the case of the instrument. Since the weight is fastened to the diaphragm *b* at its base, it is thus completely constrained except for vertical motion. The stud has an enlarged head, the underface of which serves to limit the downward travel of the weight. This stud carries a small platinum-iridium disc, *e*, on top, which makes contact with the lower face of an insulated steel plate, *f*, secured by a cap, *g*, that screws on the case. The total travel of the weight is adjusted by mica washers, and is limited to 0.001 in.

Fig. 2 shows the exterior of this instrument.

FIG. 2—EXTERIOR OF THE DIAPHRAGM TYPE OF CONTACT ACCELEROMETER

This Illustration Shows the Saddle-Piece *L* That Is Used for Static Calibration of the Instrument and the Binding Posts to Which the Telephone Watch-Case Receiver That Is Used To Indicate the Breaking of the Contact between the Small Platinum-Iridium Disc *e*, Fig. 1, and the Insulated Contact Disc *f* Is Connected through Wires

The insulated steel plate *f*, Fig. 1, has two holes, *h* and *i*, opposite recesses *j* and *k* in the stud *d*. This enables the diaphragm to be loaded, by a small saddle-piece, *L*, Fig. 2, which is used only in calibration, with known external weights acting on the inertia element *a*, Fig. 1, for the purpose of static calibration. The calibration consists of putting on known loads and adjusting the air pressure until contact is just made, or admitting an excess of air pressure and then reducing it slowly until the contact just breaks. Fig. 3 shows the static calibration curve for a given combination of diaphragm and weight.

Air pressure can be supplied from any convenient source. To obtain fine pressure-adjustments, a metallic bellows actuated by a thumb-screw is used. A liquid manometer, or other form of pressure-gage, may be used. In obtaining the curves shown herein, a manometer was used, for maximum accuracy. For observing the contact opening, a telephone head-set was generally used.

MICROMETER TYPE OF CONTACT ACCELEROMETER

Fig. 4 shows the constructional details and Fig. 5 the external appearance of a single-cell adjustable accelerometer

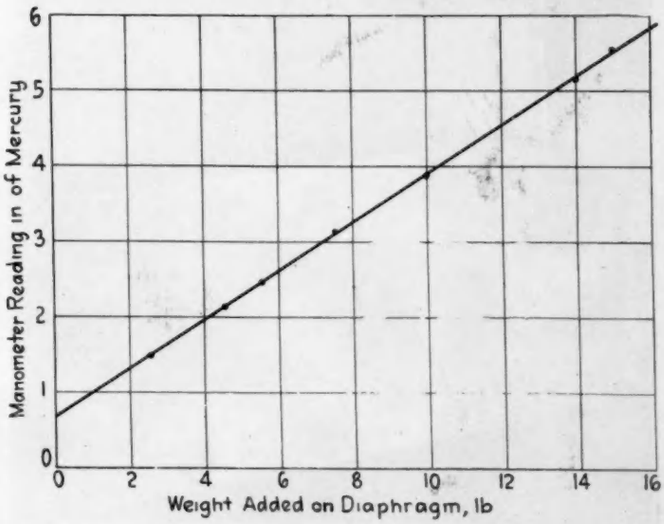


FIG. 3—STATIC CALIBRATION CURVE OF A DIAPHRAGM-TYPE CONTACT-ACCELEROMETER

This Curve Is for an Inertia Element Weighing 2.77 Lb. and the Indicated Acceleration Is Equal to the Product of the Added Weight and the Acceleration Due to Gravity Divided by the Weight of the Inertia Element. A Manometer, Reading in Inches of Mercury, Is Used as a Pressure Gage, This Type Giving the Maximum Accuracy

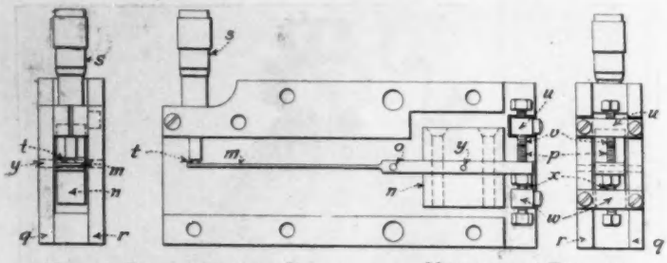


FIG. 4—CONSTRUCTIONAL DETAILS OF A MICROMETER TYPE OF CONTACT ACCELEROMETER

This Instrument Consists Essentially of a Flat Spring, *m*, with Two Rectangular Blocks That Form the Inertia Element *n*. A Contact Screw, *p*, at the Outer Extremity of the Spring Makes Contact with the Screw *v* in the Block *u*. Motion of the Screw *p* Is Limited to 0.001 In. by the Screw *x*

of the micrometer type. It consists of a flat spring, *m*, Fig. 4, with an enlarged shank to which two rectangular blocks are symmetrically bolted to form the inertia element *n*. Near the junction of spring and shank a hole is drilled to receive the pivot *o*. At its outer extremity the shank carries a contact screw, *p*.

The case or housing consists of two side-plates *q* and *r* screwed and doweled to spacer blocks running along the top and bottom sides of the plates. A standard micrometer-head, *s*, and *S*, Fig. 5, reading direct to 0.001 in. is solidly clamped at one end of the upper spacer-block, with its spindle arranged to bear directly on a hardened spherical piece *t*, Fig. 4, set in the end of the spring *m*. The side-plates are recessed to receive small blocks, the upper one of which, *u*, and *U*, Fig. 5, is insulated and carries a contact screw *v*, Fig. 4, making contact with the screw in the end of the spring shank; the lower block, *w* and *W*, Fig. 5, carries a screw, *x*, Fig. 4, that acts as an opposing stop and is adjusted to limit the total motion to 0.001 in.

The spring shank is drilled laterally at a point corresponding to the center of gravity of the inertia element *n*, and the side-plates have larger holes, *y* and *Y*, Fig. 5, on the same axis. By inserting a pin that is used only in calibration in this hole in the shank and extending beyond the side-plates, a stirrup can be hung onto the inertia element, which in turn may be loaded with pan and weights for the purpose of static calibration. This calibration consists of putting on known weights and adjusting the micrometer screw until the contact just "makes" or just "breaks." Fig. 6 shows the static calibration-curve. The sensitivity of the particular combination of weight and spring used is 5.15 ft. per sec. per sec. per 0.001 in. on the micrometer.

The inertia element weighs about ¼ lb., and the entire instrument about 3½ lb. The over-all dimensions, exclusive of micrometer thimble, are approximately 6 in. long by 2¾ in. high by 1 in. thick.

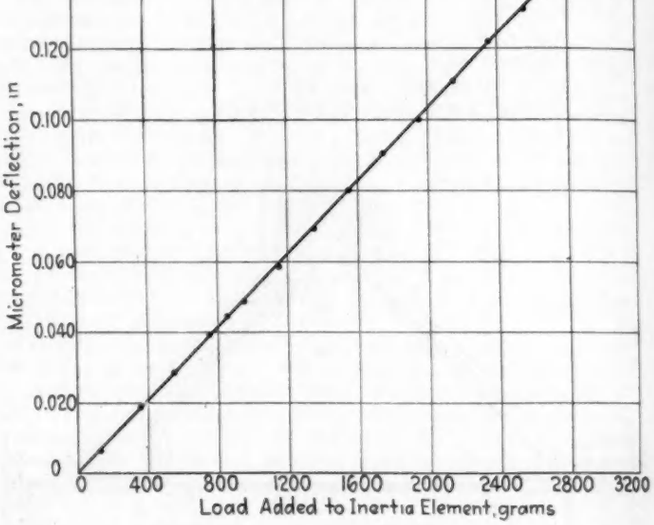


FIG. 6—STATIC CALIBRATION CURVE OF A MICROMETER-TYPE CONTACT-ACCELEROMETER

The Weight of the Inertia Element of This Instrument Is 121 Grams and the Sensitivity of the Particular Combination of Weight and Spring Used Is 5.15 Ft. per Sec. per Sec. per 0.001 In. on the Micrometer

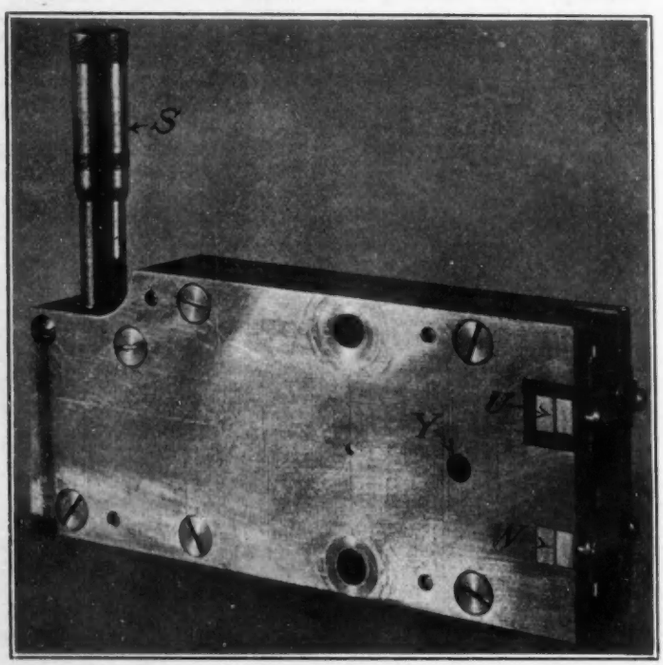


FIG. 5—EXTERIOR OF THE MICROMETER-TYPE CONTACT-ACCELEROMETER

The Micrometer *R* Provides a Means for Calibrating the Instrument by Turning the Micrometer Screw until the Contact Screws *p* and *v*, Fig. 4 Either Just Make or Break Contact When the Inertia Element Is Loaded by Placing Known Weights on a Stirrup That Is Attached to a Pin Inserted in the Hole *Y*

For use in battery construction, a design of accelerometer cell is under consideration which will weigh less than ½ lb., with a 2-oz. inertia element, and will have correspondingly small over-all dimensions. The size of the present instrument was influenced to some extent by the desire to use a standard micrometer-head. Cells for battery construction need not be adjustable, after once having been set; they require no micrometer, and can therefore be made much smaller.

DYNAMIC TESTS

Contact accelerometers are "absolute," in the sense that they can be directly and accurately calibrated against dead weights and do not require any other calibration. However, as a check on the static calibration and on the proper functioning of contacts and the like, a dynamic test is desirable.

The Firestone Engineering laboratory made available for checking purposes a filing machine that had been installed as a means for providing harmonic motion of known amplitude and frequency and hence known acceleration. While not a precision device, this furnishes a means of obtaining at least a fairly approximate check between static and dynamic calibration. The agreement obtained with the micrometer type was satisfactory, within about 1 per cent. When first mounted on the filing machine the diaphragm instrument showed a difference of about 7 per cent from its static calibration. This discrepancy was traced to the distortion of the filing machine mechanism and supports, due to the greater weight of the latter instrument, which was 10½ lb. against 4 lb. for the micrometer instrument. When the weights were equalized by adding 6½ lb. to the mounting

of the latter, the two instruments *checked each other* very closely.

Fig. 7 exhibits curves obtained in the dynamic tests.

Influence of Contact Opening

The electric-contact method of determining the separation of weight and stop is considered the best, provided the observations depend on the "break," rather than the "make," because the time-interval between imposing an acceleration in excess of that for which the cell is set and the opening of the circuit is exceedingly small and is measurable in millionths of a second, assuming, of course, that the contacts function properly. On the other hand, for measuring repeated shocks of high frequency, prompt recovery of the instrument, or closing of the contact again, is essential. If, however, successive shocks are due to periodic reversals of motion, as is the fact in a vast majority of cases, the reversal of motion automatically closes the contact and the instrument will operate at practically any frequency, provided the successive displacements are at least as great as the contact opening, and provided there is no elastic rebound of the contacts.

If successive shocks occur without reversal of motion, a condition which seldom if ever exists in practice, the periodicity of the instrument becomes important. The time required is a function of the contact-opening and the acceleration for which the cell is set, being longer for lower accelerations. Consider, for example, a cell set for 0.1 *g* or 3.2 ft. per sec. per sec., which is about as low as would ever be measured on vehicles, and let the contact opening be 0.0001 ft. or 0.0012 in. Assume that contacts have been opened by an acceleration greater than 0.1 *g*, and that the acceleration then becomes zero instantaneously; using the formula $s = \frac{1}{2}at^2$, the time required for recovery is approximately 0.008 sec. Bearing in mind that for higher accelerations the time is correspondingly shorter, and that, generally speaking, the higher frequencies are associated with higher accelerations, and bearing in mind also that contact-openings much smaller than 0.001 in. may be used in practice, it becomes clear that such accelerometers are, or can be made, rapid enough for all frequencies encountered in automotive work.

Applications

The most immediate use of such accelerometers is, of course, the measurement of vertical shocks on automobile bodies and running-gears. Other applications, however, should not be overlooked; for instance, by the simple expedient of turning these instruments on end, which changes the calibration by 1 *g*, they are immediately available for measuring longitudinal shocks on bodies or on axles; or lateral shocks. Then there is the assortment of impacts encountered in the railroad field, an investigation of which by such accelerometers might react to the advantage of the traveling public.

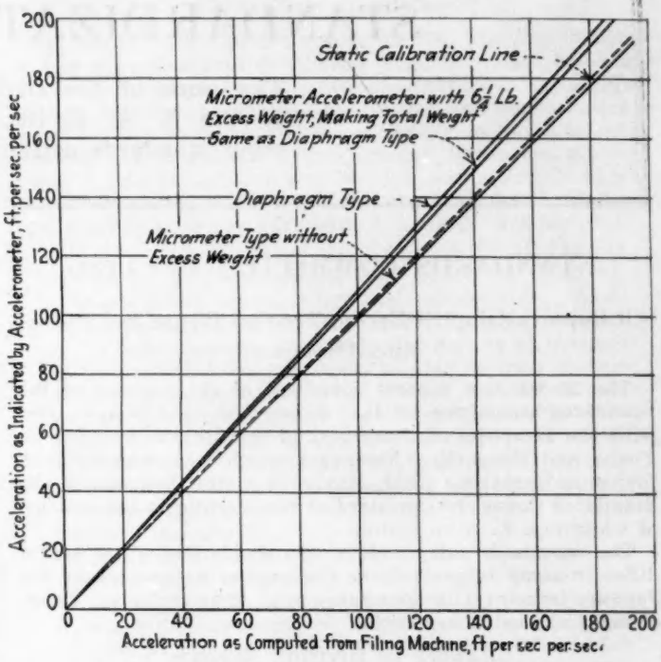


Fig. 7—Results of Dynamic Tests on Micrometer and Diaphragm Types of Contact Accelerometer

These Tests Gave a Check on the Static Calibration of the Instruments and also on the Proper Functioning of the Contacts and Other Parts. The Close Agreement of the Four Different Curves Should Be Noticed

Thanks are due to the Bureau of Public Roads, in whose shops the parts for the micrometer accelerometer were made and assembled; to R. W. Brown and the Firestone Tire & Rubber Co. for the contacts used in this instrument, and to R. B. Day, of the Goodyear Tire & Rubber Co., who has participated by carrying out some of the static-calibration work.

MEETINGS OF THE SOCIETY

(*Concluded from p.* 247)

several hundred tests have failed to show anything seriously wrong with the procedure adopted. Involved subjects such as the relative effects of various anti-knock fuels, anti-knock dopes, altitude, compression, mixture-ratio, and cylinder action have become clearer. He sees no apparent reason why the procedure adopted is not applicable to determining the relative merits of combustion-chamber shapes, various spark-plug locations and other important considerations that are necessary to the realization of higher compression-pressure and its accompanying substantial fuel-saving. Further, he thinks the incidental information gained concerning engines of present-day compression-ratios is of no small value.

As an example of its application, Professor Roesch mentioned that the method has been used to appraise the value of anti-knock dopes or fuels and to determine the amounts necessary to produce equivalent anti-knock results. The proportional blends evidently can be determined conclusively. Refining processes can be surveyed from the anti-knock standpoint and the data can be used to aid the writing of refinery control-specifications. Inspections of many proposed anti-knock dopes by using the method described have shown their effectiveness or ineffectiveness. The method apparently is applicable either to knock suppressors or to knock inducers as shown by the several tests of this nature.

Professor Roesch remarked that, in the method as used, the effects of the personal equation are more imaginary than real. A certain technique is necessary, but check tests on unknown fuels were made repeatedly and indiscriminate change-overs of the testing engineer during the series of test-runs on any fuel have been made with the same results as though the tests were completed by one engineer. During the limited time allowed for discussion of the paper, several questions were asked and answered that pertained to details of the test procedure.

STANDARDIZATION ACTIVITIES

The work of the Divisions and Subdivisions of the S. A. E. Standards Committee and other standards activities are reviewed herein

STANDARDS COMMITTEE MEETING

All Reports Adopted Except That on Piston and Piston-Ring Oversizes

The 29 Division reports submitted at the meeting of the Standards Committee on Jan. 26 were adopted as submitted with the exception of the report of the Engine Division on Piston and Piston-Ring Oversizes which was referred back for reconsideration. C. M. Manly, first vice-chairman of the Standards Committee, presided at the meeting in the absence of Chairman E. A. Johnston.

The reports as submitted for Standards Committee action differ in some respects from the reports as printed in the January issue of THE JOURNAL, pp. 17 to 24 inclusive. These changes are noted herewith.

CHANGES IN DIVISION REPORTS

Tractor Belt-Speeds.—The 3250-ft. per min. belt-speed was omitted in the report as submitted by the Agricultural Power Equipment Division.

Spark-Plugs.—The length of the thread for the ⅝ and the 13/16-in. skirt-lengths should have been ⅜ in.

Radiator Covers.—As submitted by the Passenger-Car Division, the report was changed to specify "shutters" instead of "covers."

Motorcoach Storage-Batteries.—The heading of the third column in the table should have read "8-hr." instead of "20-hr." rate.

Monobloc Containers.—As submitted by the Storage-Battery Division, the partition thickness requirements read, "The thickness of the partition shall be not less than 3/16 in. or more than 5/16 in."

Battery Nomenclature.—A revised report was submitted by the Storage-Battery Division eliminating the term "Post Adapter"; including, under Separators, "Wood Separator with Rubber Retainer"; and revising "Terminal Connectors" to read:

Terminals:
Clamp-Lug Terminal
Taper-Thimble Terminal
Cable Terminal

Following the Standards Committee Meeting, the reports were submitted to the Council and approved for submission to the members in meeting assembled prior to a mail vote of the Society members.

In view of the valuable information brought out in the discussion of the reports on Ball-Bearing Corner Radii and Piston-Ring Oversizes, the discussion of these reports is given in full herewith.

BALL-BEARING CORNER RADII

VICE-CHAIRMAN C. M. MANLY:—Your attention is called to the statement in the report on Ball-Bearing Corner Radii as printed in THE JOURNAL that while the American Standard proposed by the Sectional Committee on Ball Bearings, which is sponsored by the American Society of Mechanical Engineers and the Society of Automotive Engineers, specifies two radii for each bearing—the maximum corner-radius of the shaft or in the housing and the minimum corner-radius for the bearing—it is the opinion of the Ball and Roller Bearings Division that for automotive practice it is unnecessary to specify both radii. The Division has, therefore, recommended that a single radius be specified for each bearing, this radius to be the maximum fillet on the shaft or in the housing that will be cleared by the bearing. If the report is adopted, there will consequently be a distinct difference between the S.A.E. Standard and the proposed American or International Standard and we should have the comments of both the manufacturers and the users before approving the report.

K. L. HERRMANN:—To specify but one radius does not seem to be logical, as the car builders would not know how to inspect the bearings. There is also a question as to the need for using a radius at all. In our design we are inclined to the use of a bevel in place of a radius and after reading this report it seems logical even to consider using a concave surface in place of a fillet.

W. L. BATT:—The round corner is preferred as it prevents the bearing from picking up material from the shaft or the housing.

W. R. STRICKLAND:—This standardization takes into account the present method of manufacturing ball-bearings here and abroad. The radii conform with those specified in the proposed International Standards, but they permit the manufacturers to round the corners in a commercial way, which eliminates the need for a special operation. I think this is a logical proposal. It represents a working arrangement with the proposed International Standards.

T. V. BUCKWALTER:—There is another point that ought to be emphasized. A bearing with a rounded corner is much easier to harden than one with a chamfer.

E. H. EHRMAN:—How does the radius, chamfer or bevel limit the width of the end face or shoulder of the bearing? This is an important point. It seems to me that some maximum distance should be specified with regard to the distance that the radius should extend from the bearing bore in order to safeguard the shoulder.

VICE-CHAIRMAN MANLY:—The vote of the Ball and Roller Bearings Division on this report was 10 "yes," 0 "no" and 4 "not voting." I understand that most of the Division members are manufacturers.

MR. STRICKLAND:—In a great many cases the car builder would not want to specify the corner radius. I think it is up to the ball-bearing manufacturer to determine what shall be used. If the ball-bearing manufacturer should go too far, it would be up to the Society to consider the matter.

MR. BATT:—The report represents a practical compromise, and suits the manufacturers reasonably well. The matter of corner radii is one of the most troublesome of the minor problems that ball-bearing manufacturers have to contend with. It is not so serious in Europe, because, as wages are not so high there they can afford to form the chamfer accurately in a separate machining operation. I know that the Division spent considerable time in determining corner radii that would seem reasonable and yet not commit the ball-bearing manufacturers to uneconomical production processes.

G. R. BOTT:—In order to do the greatest good, S.A.E. Standards cover the sizes of bearing that are used by the greatest number of people. There are about three ways of making ball-bearings. Some manufacturers chamfer the rings by a separate process. Other manufacturers use a forming-tool that provides for an excess of material on the side of the ring where the radius begins; after the rings are hardened and ground, the grinding operation takes off the excess material and the original curvature is retained. There is another method in which the radius of the ring is turned as approximately a true radius. A part of the radius is then ground off in machining the bearing. When the bearing is

finished by the grinding operation, a small arc of a large radius remains.

You will appreciate that as between these three methods of manufacturing, it was a difficult problem to get all to agree on the present standards. The manufacturers then realized that they were not getting as large a radius for the bearing as the fillet on the shaft allowed. This is the reason the "R_2," or the maximum radius on the shaft or in the housing, was selected.

In reality, the user is not interested in the forming-tool. It does not make any difference whether a large or a small radius, or a true 45-deg. bevel, is used. The ball-bearing manufacturers believe that the radii now proposed will solve their problem.

A. J. SCAIFE:—I move the adoption of the report.

[The motion was seconded and unanimously carried.]

PISTON AND PISTON-RING OVERSIZES

LEONARD OCHTMAN, JR.:—The reason for the two classifications of internal-combustion engine for the purpose of specifying standard oversizes for pistons and piston-rings is not clear. A number of truck, tractor, passenger-car, and airplane engines have been used in marine applications. The pistons and piston-rings of such engines are the same, irrespective of the application. Also, several makes of marine engine are suitable for industrial use. Therefore, the classifications given in the recommendation seem rather arbitrary.

I suggest that but one series of oversizes of pistons and piston-rings be specified for all engines up to and including 7-in. bores, which is the largest bore listed in the present S.A.E. Recommended Practice for Piston-Ring Grooves. The standard oversizes should be 0.003, 0.005, 0.010, 0.015, 0.020, 0.030, and 0.040 in. The 0.003-in. oversize for piston-rings may be omitted, as recommended.

J. F. WINCHESTER:—It has been my general experience that the ordinary cylinder, provided it were ground correctly originally, will run usually about 0.008 in. oversize and can usually be trued up without taking off more than 0.015 in., so that, instead of using the standards recommended, our company has adopted standard oversizes of 0.015, 0.030, 0.045, and 0.060 in. It would seem that the standard advocated is not logical and I move that this report be referred back to the Engine Division for reconsideration.

[The motion was seconded by Mr. Scaife.]

MR. SCAIFE:—Our experience checks with Mr. Winchester's. We find in motor-truck operation that, when cylinders will run 0.007 or 0.008 in. oversize, we can grind to 0.015 in. It is unnecessary to go to 0.020 in.

G. A. YOUNG:—What was the Division vote on this?

VICE-CHAIRMAN MANLY:—Ten "for," two "not voting" and none "against."

MR. WINCHESTER:—It seems to me that the Division has defeated the very purpose they have set forth, namely, simplification. I think the present standard is much simpler than the one recommended here.

H. M. CRANE:—Are the 0.003 and 0.005-in. oversizes for replacement work or are they factory-salvage oversizes? I can see the sense of their asking for them as a factory salvage proposition, but cannot see their need in service work.

MR. WINCHESTER:—The proposed standard would encourage the average garage to replace the piston-rings when the cylinders have been slightly worn. I have seen considerable unsatisfactory work done through encouraging that sort of practice.

If the manufacturer wants an oversize piston-ring of this size, there is nothing to prevent him from getting it, but if we encourage this throughout the industry it is natural to assume that the average garage-man, instead of reboring to the proper size, will try to doctor up the job by inserting piston-rings in cylinders that are more or less out-of-round.

R. R. KEITH:—I see no reason why 0.005, 0.010, 0.015, 0.020, 0.025, 0.030, and 0.040 in. and multiples of 0.010 in. thereafter would not meet all requirements for both pistons and piston-rings. There has been no occasion for the 0.003-in. oversize in our experience.

[Mr. Winchester's motion was seconded and carried.]

BALL-BEARING NUMBERS

In the discussion of the report on Ball-Bearing Numbers it was suggested that double-row angular-contact ball-bearings should have a special series designation so as to differentiate between the double-row annular and double-row angular-contact series in the same manner as the single-row annular and single-row angular-contact series are differentiated. The reasons for the Division action in this respect were well summarized in the following letter by H. E. Brunner received after the Standards Committee Meeting.

> It is our feeling that the policy of S.A.E. bearing standardization has always been to refrain from distinctons based purely on internal construction. When types of bearing are entirely interchangeable externally, in both dimension and function, there is no occasion to distinguished between them insofar as their numerical designation is concerned. This well-established principle covers the question of differentiating between double-row bearings of the angular type and of the radial or annular type. The two bearings are entirely interchangeable as to dimension and both serve as self-contained double-thrust carriers, in addition to their radial load capacity.
>
> On the other hand, there is a basic distinction between single-row bearings of the angular-contact and of the radial or annular-contact type. In this case, the external dimension of the bearings will vary by the amount of overlap allowed for the races and, in addition, the angular-contact single-row bearing is essentially a one-direction thrust-carrier. This latter property involves a fundamental distinction that must be borne in mind in laying out the bearing application, since provision must normally be made for stabilizing the shaft in both directions.
>
> It is evident from the above considerations that it is undesirable to make any distinction between double-row bearings in the matter of angle of contact. If such a distinction were made, it would be equally logical to attempt to differentiate between a 20-deg. and a 40-deg. contact-angle.
>
> We believe that the S.A.E. standardization should be kept as simple and inclusive as possible in order to provide the practical assistance it should afford to both the manufacturer and the consumer.

ROLLER BEARINGS

Following the Standards Committee action on the recommendations submitted by the Ball and Roller Bearings Division, T. V. Buckwalter, of the Timken Roller Bearing Co., presented a progress report covering the work of the Subdivision on Roller Bearings, which consisted of G. H. Adams, of the Bock Bearing Co., and Mr. Buckwalter.

The report is to be submitted to the Division for action at its next meeting and will probably be submitted to the Standards Committee at the Summer Meeting.

RADIATOR NOMENCLATURE REVISED

Evaporative Cooling Not to Be Covered Until Method Is in Commercial Use

The following definitions for radiator core were recommended for adoption as S.A.E. Standard Nomenclature at the Jan. 25 Meeting of the Radiator Division held in Detroit at the Book-Cadillac Hotel:

> *Individual Fin and Tube Core*.—An assembly of fluid tubes of any cross-sectional form to each of which are attached gills or fins of circular, square or other shape, each tube and its fin or fins forming a separate unit.
>
> *Continuous Fin and Tube Core*.—An assembly of fluid tubes of any cross-sectional form, the tubes being joined together by radiating fins or plates common to all tubes or groups of tubes.
>
> *Ribbon Cellular Core*.—A number of fluid passages made by joining metal ribbons at the edges and

grouped to form a cellular structure. Parts of the cellular structure may be of formed or flat ribbon which is not a part of the fluid passage.

Air Tube Cellular Core.—An assembly of air tubes nested in such a way as to form fluid passages between the tubes, the passages being sealed at the ends of the tubes. In this type the fluid may flow transversely as well as vertically around the tubes.

These definitions were originally submitted by the Radiator Division for action at the June, 1923, meeting of the Standards Committee, but were referred back to the Division owing to objections raised at that time. These objections were carefully studied at the January Meeting and revisions made such that it is believed that the revised definitions will meet now with general approval.

The present S.A.E. Standard nomenclature covering radiators of the shell and cast type, as given on p. K5a of the S.A.E. HANDBOOK, was also revised, being extended to give a separate classification for the "core and tank assembly," the term "core" being very often used incorrectly when it is desired to designate the complete "core and tank assembly." The nomenclature as revised follows:

SHELL TYPE

Radiator Core and Tank Assembly

Radiator core
Radiator core header sheets
Radiator upper tank
Radiator filler-neck
Radiator filler-neck sleeve
Radiator filler-cap
Radiator filler-cap gasket
Radiator tie-rod fitting
Radiator baffle
Radiator inlet fitting
Radiator lower tank
Radiator outlet fitting
Radiator drain flange
Radiator drain-cock
Radiator anchor stud or bolt
Radiator anchor stud or bolt plate
Radiator overflow tube
Radiator side bolting-member
Radiator shell anchorage clips

Radiator Shell

Radiator supports
Radiator anchor studs or bolts
Radiator support reenforcement
Radiator hinge-rod fitting
Radiator brace-rod fitting
Radiator hood-ledge liner strip
Radiator starting-crank hole cover

CAST TYPE

Radiator Core Assembly

Radiator core
Radiator core upper header
Radiator core lower header
Radiator core overflow jacket tube

Radiator Upper Tank

Radiator filler-cap
Radiator filler-cap gasket
Radiator filler-cap hinge-pin
Radiator filler-cap fastener
Radiator tie-rod fitting
Radiator hinge-rod fitting
Radiator inlet fitting
Radiator inlet gasket
Radiator inlet studs or cap-screws

Radiator Lower Tank

Radiator anchor studs or bolts
Radiator outlet fitting
Radiator outlet gasket
Radiator outlet studs or cap-screws
Radiator drain-cock or plug
Radiator clamping strips
Radiator clamping bolts or studs
Radiator overflow tube
Radiator sides
Radiator header gasket
Radiator hood-ledge liner strip

Although it had been suggested that nomenclature be drawn up to cover the evaporative method of cooling, this was decided inadvisable until the evaporative method of cooling is in commercial use.

Those present at the Division meeting were J. D. Harris, of the McCord Radiator & Mfg. Co., chairman; Charles Oppe, of the G & O Mfg. Co.; W. C. Ritter, of the Long Mfg. Co.; L. P. Saunders, of the Harrison Radiator Corporation; F. M. Young, of the Racine Radiator Co. and C. E. Heywood, of the Society's Standards Department.

S.A.E. CASTLE-NUT STANDARD REVISED

Screw-Threads Division Recommends Adoption of Sectional-Committee Dimensions

E. H. Ehrman, of the Chicago Screw Co., and O. B. Zimmerman, of the International Harvester Co., have submitted to the Screw-Threads Division a report covering the present S.A.E. Standard for Castle Nuts, printed on p. C3 of the S.A.E. HANDBOOK, revised in accordance with the progress report of the Sectional Committee on Bolt, Nut and Rivet Proportions, organized under the rules of procedure of the American Engineering Sectional Committee.

The revised dimensions, which are given in the accompanying table, are in accord with the present S.A.E. dimensions with the exception that

The diameters across the flats for ¾-in. and larger diameters have been increased to conform to the Sectional-Committee Specifications for finished nuts.

The maximum widths across the corners have been taken from Tables 6 and 8 of the Sectional-Committee Specifications.

The width-across-corners dimension serves two purposes; it determines the minimum width across the flats and the minimum distance between the bearings of the nut in the wrench opening.

The report will probably be submitted to the Standards Committee for action at the June meeting.

PROPOSED DIMENSIONS FOR S. A. E. STANDARD CASTLE NUTS

Diameter	Threads per Inch, NF-2	Width Across Flats, Nominal[1]	Width Across Corners, Minimum	Thickness, Nominal	Slot Width	Slot Depth
1/4	28	7/16	0.488	9/32	5/64	3/32
5/16	24	1/2	0.557	21/64	5/64	3/32
3/8	24	9/16	0.628	13/32	1/8	1/8
7/16	20	5/8	0.699	29/64	1/8	1/8
1/2	20	3/4	0.840	9/16	1/8	3/16
9/16	18	7/8	0.980	39/64	5/32	3/16
5/8	18	15/16	1.051	23/32	5/32	1/4
3/4	16	1⅛	1.263	13/16	5/32	1/4
7/8	14	1 5/16	1.474	29/32	5/32	1/4
1	14	1½	1.686	1	5/32	1/4
1⅛	12	1 11/16	1.898	1 5/32	7/32	5/16
1¼	12	1⅞	2.109	1¼	7/32	5/16
1½	12	2¼	2.533	1½	1/4	3/8

[1] The nominal width across flats is also the maximum width.

Vol. XVIII March, 1926 No. 3

High Average Operating-Temperature and Engine and Car Operation

ANNUAL MEETING PAPER *Illustrated with* CHARTS AND DRAWINGS

ABSTRACT

THIS subject is treated in a paper in two parts. Part I, by Alex Taub, deals with laboratory tests to prove by comparative data that the higher average operating-temperatures maintained in the engine by the constant-temperature, or evaporation, system of cooling have negligible detrimental effects. Part II, by L. P. Saunders, gives the results of road-tests of cars operated under the same conditions when fitted with a standard water-cooling radiator-core and with a constant-temperature cross-flow condenser-core.

Although contamination of the crankcase oil by heavy ends of the fuel is not prevented by the higher temperature of constant-temperature operation, it is asserted that this higher temperature is effective in striking an acceptable balance in such contamination and results of the tests show that the cylinder-walls are maintained at temperatures sufficiently above the vaporization point of water to reduce the condensation of water vapor to the minimum. Water in the crankcase is the objectionable element. Oil dilution by fuel up to a certain amount is not detrimental; in fact, experience shows that about 16 per cent of such dilution is necessary to facilitate starting a cold engine. Even when an anti-freeze solution containing 50 per cent of alcohol is used and the boiling temperature reduced to 184 deg. fahr., the cylinder-wall temperatures are maintained at 212 deg. or more.

Since a boiling liquid does not change its temperature, it affords the simplest means of maintaining a constant operating-temperature and also the simplest, least expensive and lightest means of providing for quick warming-up of the engine and slow cooling-down, because there is no circulation of water except when steam is passing from the engine-block to the radiator or condenser.

Test runs were made in the laboratory with an engine fitted with a Muir constant-temperature system which could be converted to water-cooling by blocking-off the circulation through the cylinder-head with a special gasket to provide for concentrated circulation around the exhaust-valves. Outlet-water temperatures were controlled by admitting more or less cold water. Results of the tests indicate that fuel consumption is approximately the same for constant-temperature cooling at 212-deg. outlet temperature and water-cooling at 170-deg. outlet temperature; that with both systems the spark-lever advance for maximum torque is safely below the degrees of advance before detonation, or spark knock, begins; that the falling-off in torque with reduction in richness of the fuel mixture is virtually parallel for the two systems; that the difference in volumetric efficiency of the engine when operated on the two systems amounts to only 2 per cent, which is within the allowable error of the air-meter used; that the brake engine-pull is nearly identical; that the temperature of the lubricating oil is not affected by the system of cooling but by the temperature of the cooling medium, and that the temperature of the walls and inlet and exhaust-valve seats of the No. 1 and No. 6 cylinders of a six-cylinder engine is much more uniform with constant-temperature cooling than with water-cooling.

Cylinder-head formation and spark-plug location are important factors as regards detonation. A compact head with spark-plug carefully located to allow the maximum spark-advance before detonation starts provides sufficient leeway for the use of higher operating-temperatures.

With constant-temperature cooling it is advisable that the normal water-level be such that, in operation, the water flowing from the radiator to the engine-block will fill the pipe only about half full and allow air to escape above it, thereby eliminating the possibility of an air-trap in the water-pump.

A steam-dome capacity equal to 21 per cent of the normal quantity of water in the engine-block gives the proper proportion of water and steam passing to the radiator. The smallest possible quantity of water is the proper quantity to use, as the quantity of water in the block controls the warming-up period. The normal water-level is raised between 12 and 15 per cent by expansion and volcanic action of the water when the engine is running, and the capacity of the steam dome is thereby reduced 6 or 9 per cent. If the steam dome is too small, excess water will pass to the radiator.

That the high operating-temperatures that develop with constant-temperature cooling are safe is indicated by the much higher operating-temperatures in air-cooled engines.

In Part II, after pointing out the general recognition of over-cooling by the water cooling-system in winter, as made evident by the use of air shields on the radiators, and describing the operation of the Muir cross-flow condenser-core, the author gives the results of many road-tests of cars with water-cooling and constant-temperature cooling. It is shown that the cross-flow core, when used as a water-cooler, maintains a lower temperature of the outlet water from the radiator than the conventional core and still lower temperatures when used as a constant-temperature system.

Miles per gallon of fuel consumed are increased by constant-temperature cooling with the cross-flow core as compared with water-cooling with the standard core. Acceleration tests showed a slight advantage for the former system, while deceleration times were slightly longer than with water-cooling. The constant-temperature system showed higher speeds in hill-climbing.

Better ventilation of the engine hood may be necessary with the constant-temperature system to avoid uncomfortable heat in the driving portion of the car body. Size of the radiator-core cannot be decreased, as many cars are now inadequately cooled under certain extreme conditions of driving and air temperature and density. The fan size should remain as large, at least, as at present.

The paper is concluded with a chart showing the effects of temperatures in the water-jacket and in the lower part of the radiator caused by starting and stopping of the car in cold weather.

PART I BY ALEX TAUB[1]

ASSUMING that the general principle of constant-temperature cooling, or steam-cooling, is understood and that the idea no longer exists that the cylinder-block is full of steam instead of water, an attempt will be made to prove by test data the negligi-

[1] M.S.A.E.—Development engineer, General Motors Corporation, Detroit.

ble extent of the detrimental effects of the higher average-temperature maintained by this system of cooling as compared with the customary water cooling-system. Some of the charts have been presented before, but they are essential to the exposition of the theme.

The higher temperature of the constant-temperature system is, naturally, not a cure-all for crankcase-oil dilution. There can be no doubt, however, of its great value in effectively reducing the condensation of water to the desirable minimum and also in striking an acceptable balance of contamination of the lubricant by condensed heavy ends of the fuel. Should complete prevention of such contamination be effected, the starting of a cold engine would become a serious problem. It is our experience that it is necessary to maintain about 16 per cent of contamination, provided there is practically no water in the oil mixture. Water is the unwelcome element in the crankcase. We have been unable to discover any useful purpose that is served by water in this location. Accumulation of water there is held down effectively by the higher operating-temperature maintained by the constant-temperature cooling-system. An analysis of how water accumulation is affected by the temperatures of the metal of the engine during operation follows.

Relation of Engine Temperature to Oil Dilution

All engine-block temperatures were taken by thermocouples inserted in the metal of the block to within 1/16 in. of the inner wall, hence the temperatures given should be fairly representative, particularly since they represent the averages of many readings.

TABLE 1—TEMPERATURES OF METAL OF WATER-COOLED ENGINE

Outlet-Water Temperature, Deg. Fahr.	Combustion-Chamber Temperature, Deg. Fahr.	Cylinder-Barrel Temperature, Deg. Fahr.
110	145	146
120	153	156
130	162	167

Temperatures of the combustion-chamber of an L-head engine, running with full load at 1000 r.p.m., taken over the exhaust-valves at the rear of the block, and of the cylinder-barrel metal, taken on the no-valve side within ½ in. of the top, were indicated as in Table 1. The water temperatures were taken at the water outlet and would be considered fairly warm for winter operation.

It is obvious that it is difficult, with these temperatures, to maintain vaporization of the water vapor that is a by-product of combustion. When it is considered that the volume of this by-product, when condensed, is equal to approximately 1⅛ times the volume of the fuel burned, it becomes evident that, should only 10 per cent of the water vapor be condensed, possibly 2 qt. of water could be accumulated in 100 miles of car operation, assuming a fuel consumption of 1 gal. to 20 miles. This is worth thinking about.

For comparison with the foregoing temperatures of the metal, the temperatures taken at the same points but with a water-outlet temperature of 170 deg. fahr., which represents the desirable operating-temperature for water-cooling, are submitted in Table 2.

Boiling Fluid Maintains Constant High Heat

These temperatures are an aid in maintaining vaporization of the water vapor, but a water temperature

TABLE 2—ENGINE TEMPERATURES WITH OUTLET-WATER TEMPERATURE OF 170 DEG. FAHR.

	Deg. Fahr.
Top of Combustion-Chamber	199
Cylinder-Wall ½ In. from Top	207
Cylinder-Wall ½ Stroke Down	188

of 170 deg. cannot be held constant in all territories at all times of the year with any of the methods in common use today. Some improvement in general operating-conditions can be obtained by a hot intake-temperature to warm-up the metal, with a sacrifice in volumetric efficiency. However, thermal difficulties would arise long before a satisfactory reduction in vapor condensation was obtained by this method. Apparently, what is desired is a means for automatically maintaining a high constant-temperature regardless of climatic conditions.

The simplest means of maintaining a constant temperature is by a cheap fluid raised to the boiling-point which is within the operating-temperature range of the engine and which will not vary, as it is a function of a natural law that the temperature of an unconfined boiling liquid does not change. Assuming that it may be safe to operate an internal-combustion engine under the constant temperature of boiling water, it is of interest to note the temperatures the metal of the cylinders has under the condition of boiling water at an outlet temperature of 212 deg., shown in Table 3. No condensation of water vapor in the cylinders should occur with these temperatures. However, as the operating-temperatures would be affected in winter by the introduction of an anti-freeze solution in the cooling medium, engine temperatures were taken with a 50-per cent alcohol solution which had a boiling-point of approximately 184 deg. fahr. and were found to be as given in the last column of Table 3. It is patent that these operating-temperatures will effect considerable reduction in condensation.

Should Develop High Temperature Quickly

To derive full advantage of the benefits, if any, to be obtained from high constant operating-temperatures, it is necessary that these temperatures be reached as soon as possible. Hence it is clear that the temperature-controlling medium should not be cooled while an attempt is being made to heat it, as is the case with the majority of water-cooled cars. Quick warming-up of the water can be obtained in several ways by mechanical means, but the simplest method is to arrange for the minimum quantity of cooling fluid to heat and to have it positively static within the block during the heating-up period. An additional desirable condition would be to retain the heat in the engine as long as possible after the engine has been stopped. This can be accomplished to some extent by mechanical means, but the simplest way would be to eliminate the thermo-siphonal action, which occurs with customary water-cooling, while the engine cools off, which it does rapidly in cold weather.

The condition of constant high-average temperature, arrived at quickly and maintained for the longest period after the engine has stopped, is met best, with the

TABLE 3—ENGINE TEMPERATURES WITH BOILING FLUID

Outlet Temperature, deg. fahr.	212	184
Top of Combustion-Chamber, deg. fahr.	242	218
Cylinder-Wall, ½ In. from Top, deg. fahr.	246	226
Cylinder-Wall, ½ Stroke Down, deg. fahr.	237	212

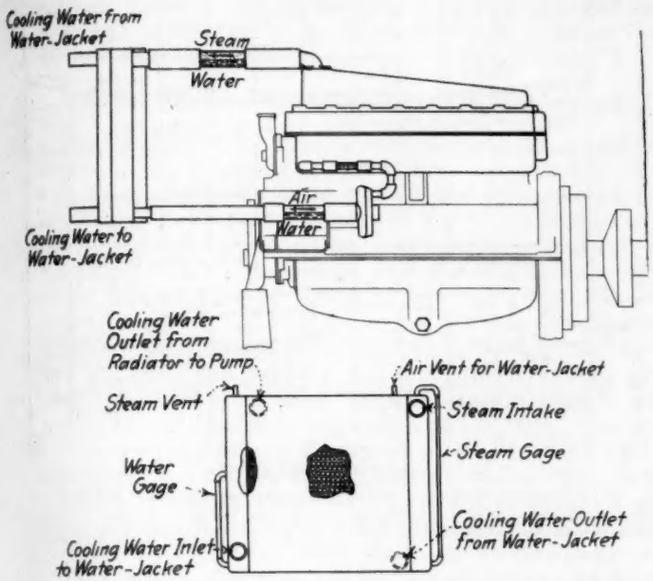


FIG. 1—SET-UP OF MUIR CONSTANT-TEMPERATURE COOLING-SYSTEM FOR TEST PURPOSES

The Pump Is Set with Its Axis ½ In. above the Bottom of the Radiator so That the Return Water Only Half Fills the Return Pipe. This Permits Air To Escape Above the Water and Prevents an Air-Trap in the Pump. The Lower Drawing Shows Positions of the Radiator Inlets and Outlets and Steam and Air Vents

minimum of expense and weight and in the simplest form, by the evaporation system, or vapor-cooling, or, if you will, steam-cooling.

TESTS TO DETERMINE POSSIBLE DISADVANTAGES

It remains to be seen whether or not the higher average operating-temperature maintained by evaporation-cooling involves any mechanical or functional disadvantages. To determine this, observations were made during tests of an engine that was equipped with a Muir constant-temperature system. The engine used for the purpose was a six-cylinder L-head engine, with 2¾x3¾-in. cylinders, having 169-cu. in. piston displacement. Conversions were made from one system of cooling to the other by changing the cylinder-head gasket to permit a general circulation through the cylinder-head for constant-temperature cooling and for concentrated circulation around the exhaust-valves for water-cooling. To protect the principles of water-cooling from any unfair comparison, considerable time was spent in calibrating the degree of concentration until it was possible to maintain a water-outlet temperature of 170 deg. fahr. with the engine running at 1000 r.p.m. with throttle full open and with only a 5-deg. difference between the inlet and outlet-water temperatures. This low difference in temperatures was sought in the interest of uniformity.

For constant-temperature operation, a make-up pump was used for returning the condensate from the radiator to the engine. For water-cooling, the water was circulated by a dynamometer pump, the water temperature being varied by the addition of more or less cold water. The make-up pump was located on the left side of the engine with its center-line ½ in. above the bottom of the radiator, as shown in the upper drawing in Fig. 1.

The cylinder-head used in all tests was a heart-shaped combustion-chamber with a spark-plug location that was determined by careful research. The water holes in both the cylinder-block and cylinder-head were of sufficient size to permit general circulation for constant-temperature operation and were blocked off by a special gasket for water-cooling.

Each run was made in a way to bring out the fact as to whether or not a particular result was affected adversely by the high-average temperature of the constant-temperature cooling method. The following specific tests were made:

(1) Full-horsepower run, to determine torque economy, at 170 deg. water-cooling outlet-water temperature and at 212 deg. constant-temperature outlet-vapor temperature
(2) Effect on spark-advance and fuel economy at 600 and 1000 r.p.m., at the foregoing temperatures
(3) Effect on mixture-ratio, as shown by torque and economy, at 600 and 1000 r.p.m., at the foregoing temperatures
(4) Volumetric efficiency from 200 to 2400 r.p.m., at the foregoing temperatures
(5) Volumetric efficiency at 1000 r.p.m. for varying temperatures, with water-cooled temperature variation from 110 to 200 deg. fahr. and constant-temperature variation from 184 to 250 deg., as follows: 184 deg. with a 50-per cent alcohol solution, 212 deg. with pure water, 225 deg. with a 32-per cent calcium chloride solution, and 250 deg. with a 56-per cent calcium chloride solution
(6) Effect of temperature on brake pull and indicated pull, with water-cooled temperatures of from 110 to 200 deg. and constant temperatures from 184 to 250 deg.
(7) Proportion of steam dome to water content
(8) Temperature of iron of cylinder-head block at cooling temperatures of from 110 to 250 deg. fahr.

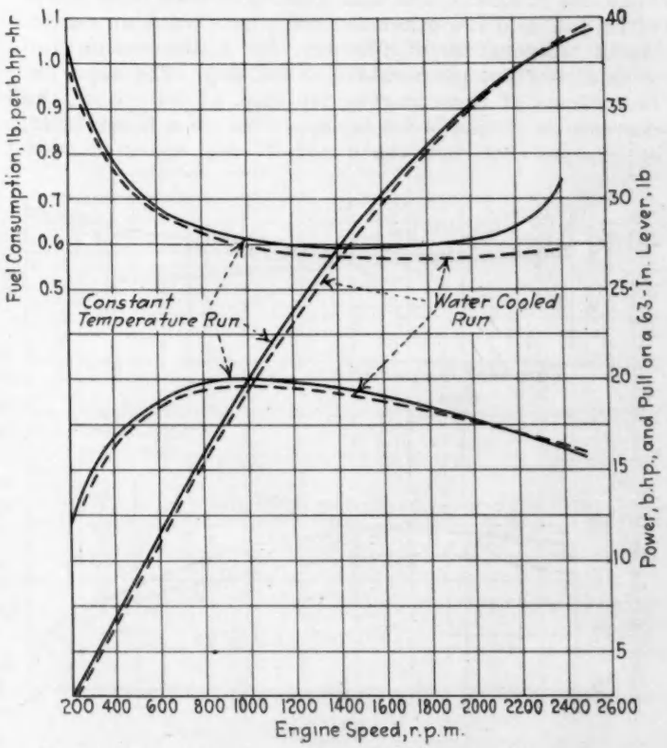


FIG. 2—BRAKE-HORSEPOWER AND FUEL-CONSUMPTION CURVES WITH WATER-COOLED AND CONSTANT-TEMPERATURE OPERATION

The Outlet Temperature for Water-Cooling Was 170 Deg. Fahr. and That for Constant-Temperature Cooling Was 212 Deg. Fahr. The Curves Show ½-Hp. Advantage for the Latter System at Speeds between 1000 and 1600 R.P.M. The Economy Curves Were Taken at Full Load with Wide-Open Throttle and Indicate Very Little Difference in Fuel Consumption between the Two Systems

Comparative Fuel Consumption and Torque-Drop

The full power and economy curves in Fig. 2 are corrected for barometric pressure and room temperature down to 29.92 in. of mercury and 60 deg. fahr., respectively. The correction for the water-cooled operation-curve, as shown, is 4.8 per cent and that for the constant-temperature operation-curve is 4.3 per cent. The temperature for water-cooling was 170 deg. and that for constant-temperature cooling was 212 deg. The curves apparently show ½-hp. advantage at between 1000 and 1600 r.p.m. for the latter system. The economy curves were taken at full load with throttle wide open and indicate that there is very little difference in fuel consumption between the two systems, the comparative figures being given in Table 4.

TABLE 4—COMPARATIVE FUEL CONSUMPTION

Engine Speed, R.P.M.	Water Cooled at Constant Temperature of 170 Deg. Fahr. Consumption Per Brake Horsepower-Hour, Lb.	212 Deg. Fahr. Consumption Per Brake Horsepower-Hour, Lb.
600	0.685	0.675
1000	0.610	0.620
1500	0.575	0.600

Curves shown in Figs. 3 and 4 represent the decrease in torque and the effect on fuel consumption due to various spark-lever positions. The scale used for these curves necessarily is broad, to show more clearly the torque decrease characteristics. The tests were made by running the engine at maximum torque for each speed and then varying the spark-lever position both ways from this mark and measuring the decrease. Two runs are indicated for each cooling-system, one set at 600 r.p.m. and the other at 1000 r.p.m., with an outlet-water temperature of 170 deg. for water-cooling and with a constant temperature of 212 deg. The apparent indications of these curves are that, at 600 r.p.m., the decrease in torque is 0.3 lb. measured on a 63-in. lever, or 1½ per cent, between 0 and 17 deg. spark-advance,

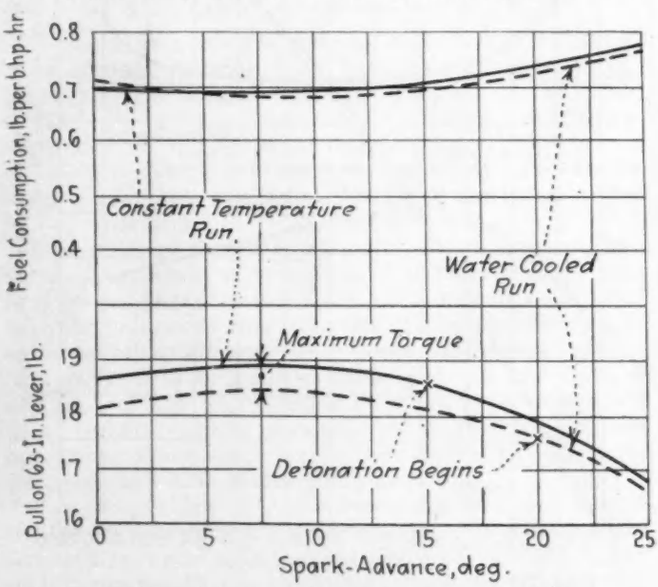


Fig. 3—Sensitiveness to Spark-Advance at 600 R.P.M.

The Fuel-Economy Curves for Constant-Temperature and Water-Cooled Operation Overlap on Even This Broad-Scale Chart. The Torque Decrease between 0 and 17 Deg. Advance Is Only 0.3 Lb. at 63-In. Lever, or 1.5 Per Cent, for Both Constant Temperature at 212 Deg. Fahr. and Water-Cooling at 170 Deg. The Operating Spark-Advance Is Safely Below the Detonating Point with Both Systems

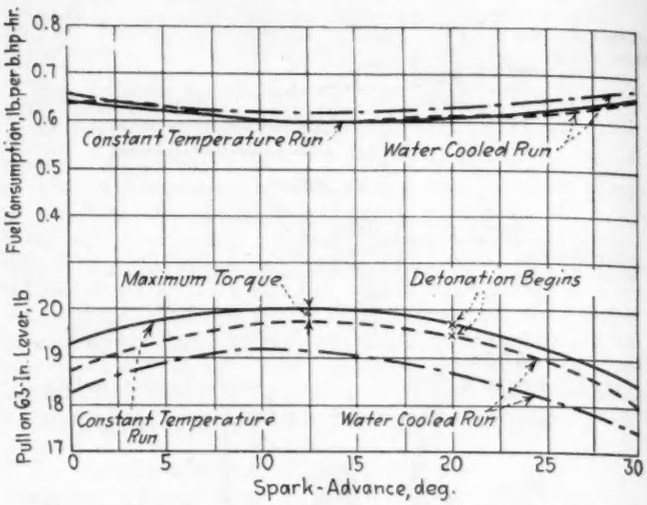


Fig. 4—Sensitiveness to Spark-Advance at 1000 R.P.M.

At This Engine Speed the Torque Decrease between 0 and 25 Deg. Advance Is 0.6 Lb. at 63-In. Lever, or 3 Per cent, with Both Cooling Systems

and, at 1000 r.p.m., the decrease is 0.6 lb. at 63 in., or 3 per cent, between 0 and 25 deg. spark-advance. These rates hold good for both water-cooling at 170 deg. and constant-temperature cooling at 212 deg. The results as to the relative effect are so close that, even with this broad scale, the economy curves overlap.

Detonation Affected by Cylinder-Head Form and Spark-Plug Location

Observations for detonation, or spark knock, as given in Table 5, indicate that in both cases the operating spark-advance is safely below the detonating point.

TABLE 5—EFFECT OF TEMPERATURE AND SPARK-ADVANCE ON DETONATION

Engine Speed, r.p.m.	600	1000
	Detonation Begins with Spark-Advance of	
	Deg.	Deg.
Water-Cooled, 170 deg. fahr.	20.0	20.0
Constant Temperature, 212 deg. fahr.	15.0	25.0
Maximum Torque	7.5	12.5

The curves in Fig. 3 indicate that the apparent detonation is not affected by the higher temperature of constant-temperature cooling. This is true in this instance; however, a water-cooled engine that is bad as regards detonation will become much worse if it is converted to constant-temperature cooling. The satisfactory results here indicated were due entirely to the compact cylinder-head and the spark-plug position shown in Fig. 5. When comparison is made between a broad flat combustion-chamber and a compact chamber, a definite shortening of the spark-advance range for maximum torque is noticed. If, in addition to a compact head, the spark-plug is located in a position to obtain the maximum spark-advance before detonation starts, there is sure to be enough leeway to permit of the use of the higher temperature. This problem of cylinder-head formation and spark-plug position presents an interesting story with any engine, whether water-cooled or not, particularly as regards hot-weather operation. Fig. 6 is submitted as an interesting comparison of the spark-advance characteristics of an engine fitted with a semi-compact and a compact head. Note the relative positions of the maximum-torque spark-advance and the detonating spark-advance

curves. These offer something to think about when plotting comparative horsepower-curves and mean-effective-pressure curves.

Torque Effects with Varying Air-Fuel Ratios

Curves shown in Figs. 7 and 8 serve as a further means of studying the effects of high-average temperature upon the sensitiveness of engine operation. They indicate the decrease in torque for varying mixture-ratios for both water-cooled and constant-temperature operation. The water-cooled test was run at an outlet-water temperature of 170 deg. fahr., and the constant-temperature test at 212 deg. The runs were made by setting the carbureter for maximum torque for each engine speed and varying the mixture-ratio, both rich and lean, and measuring the decrease in torque and the effect on the fuel consumption.

The curves are practically parallel, and the indications are that, at 600 r.p.m., varying the mixture from 8.5 lb. of fuel per hr. to 6.5 lb., a reduction of 24 per cent, there is a torque loss for the water-cooled engine of 0.25 lb. at a 63-in. lever-length, or 1.6 per cent, and a torque loss for the constant-temperature operation

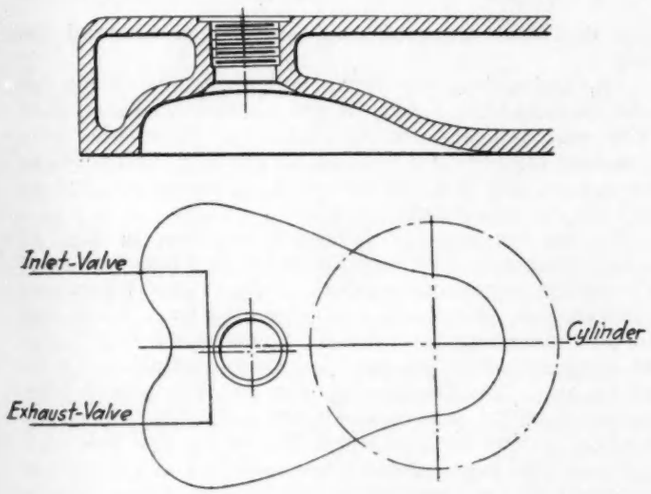


Fig. 5—Compact Cylinder-Head and Studied Spark-Plug Location
The Spark-Plug Is Located To Give Maximum Spark-Advance with Maximum Torque Before Detonation Begins and Allows Ample Leeway for the Use of the Higher Operating-Temperature of the Constant-Temperature System

of 0.60 lb., or 3.2 per cent. This is a difference of 1.6 per cent between the two systems. Inasmuch as there is a difference of 2 per cent in magnitude between runs in favor of constant-temperature operation, which difference is negligible in itself, it apparently is safe to consider a difference of 1.6 per cent in the torque decrease as negligible.

At 1000 r.p.m., it is found that varying the mixture-ratio from 14.5 lb. per hr. to 11.5 lb., or 24-per cent reduction, the torque loss with water-cooling at 170-deg. outlet temperature is 0.25 lb. at 63-in. lever-length, or 1.6 per cent, and the torque loss with constant-temperature operation is also 0.25 lb. at 63 in., or 1.6 per cent, hence there is no difference between the effects of the two systems, which is exactly the point that it is desired to bring out. Incidentally, considerable might be said with regard to this method of charting engine characteristics. For instance, our experience shows that the curves indicate that the external distribution of the engine used was not so good as it might be. That, however, is an involved story.

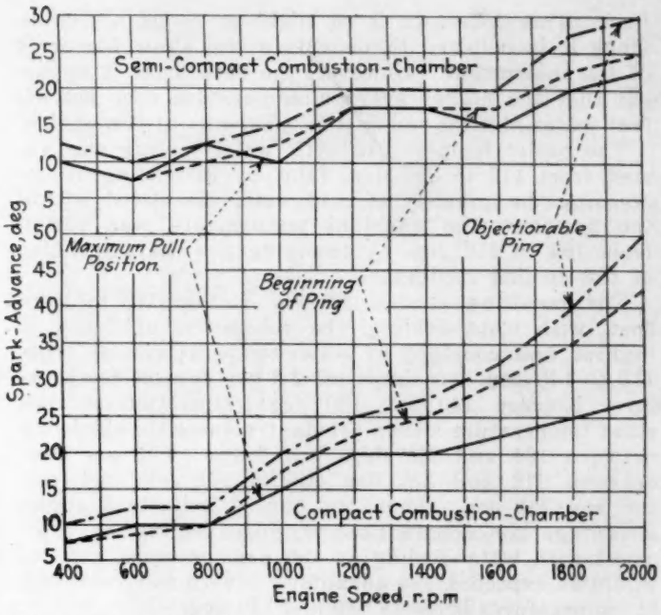


Fig. 6—Spark-Advance Characteristics with Constant-Temperature Operation with Compact and Semi-Compact Heads
Note the Relative Positions of the Maximum-Torque Spark-Advance and the Detonating Spark-Advance Curves

Effects on Volumetric Efficiency Compared

Volumetric efficiency was measured by an air-meter that measures the cubic feet of air per minute that pass through it. The engine-runs for these tests were made with water-cooling at 170 deg. and with constant temperature at 212 deg. The indications, as shown in Fig. 9, are that, from 400 to 600 r.p.m., there is a 2-per cent difference in volumetric efficiency in favor of water-cooling, and from 600 to 2200 r.p.m. there is a 2-per cent difference in favor of constant tempera-

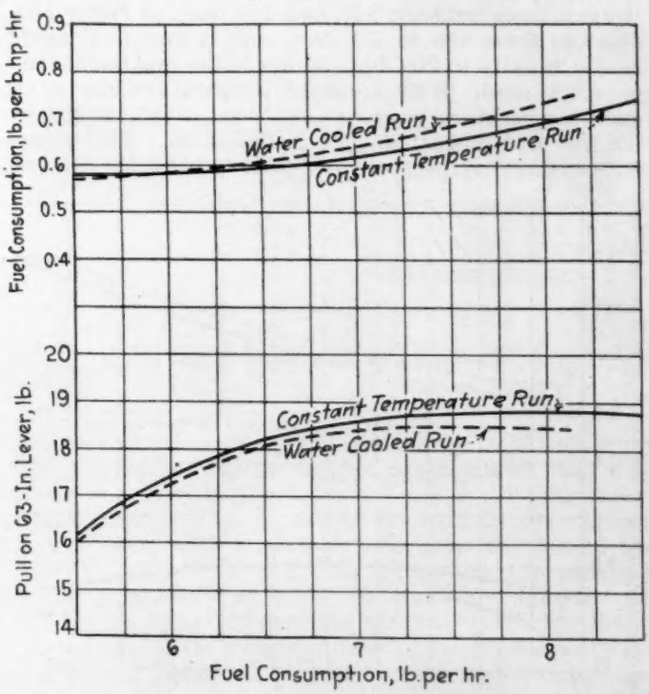


Fig. 7—Relation of Torque to Air-Fuel Ratio with Water-Cooled and Constant-Temperature Operation
At 600 R.P.M. the Curves of Torque Drop-Off with Decrease of Fuel Are Practically Parallel

ture. This difference is so slight as to be negligible, since it is believed to be within the allowable error of the instrument. However, the data make it apparent that the higher average-temperature does not affect materially the volumetric efficiency of the engine.

The outlet temperature with water-cooling was varied from 110 to 200 deg. fahr. by increasing or decreasing the quantity of cold water circulated within the system. The constant temperature was varied from 184 to 250 deg. by changing the characteristics of the cooling medium.

The resulting curves, shown in Fig. 10, indicate that, with water-cooling, the volumetric efficiency is highest and constant at water-temperatures of from 110 to 140 deg. but drops off 2.5 per cent at temperatures between 140 and 180 deg. Operation at constant temperature shows constant volumetric efficiency between 184 and 212 deg., but drops off 1 per cent between 212 and 225 deg. and 6 per cent between 225 and 250 deg. Thus, the curves indicate a slight advantage for constant-temperature cooling as compared with water-cooling at 180 deg. or more, but, as would be expected, the advantage is with water-cooling at temperatures between 110 and 140 deg.

Effect of Temperature on Engine-Pull

The curves in Fig. 11 indicate the effect on torque of water-cooling and constant-temperature cooling, and also the effect of temperature on the indicated engine-pull, which takes into consideration the relative friction. A run was made at 1000 r.p.m. and the temperature for water-cooling was varied by changing the quantity of cold water circulated in the system. Changes in temperature for constant-temperature operation were made by varying the cooling medium, as previously described. The broken lines, which indicate results obtained with water-cooling, show that the actual brake-pull increases at temperatures between 110 and 130 deg., then decreases rather rapidly at temperatures between 130 and 160 deg., is fairly constant at from 160 to 170 deg., and it drops off again rather rapidly to 200 deg., which is the maximum temperature used. With constant temperature, the minimum temperature used was 184 deg., accomplished by the use of a 50-per cent alcohol solution. The torque for constant temperature is constant between 184 and

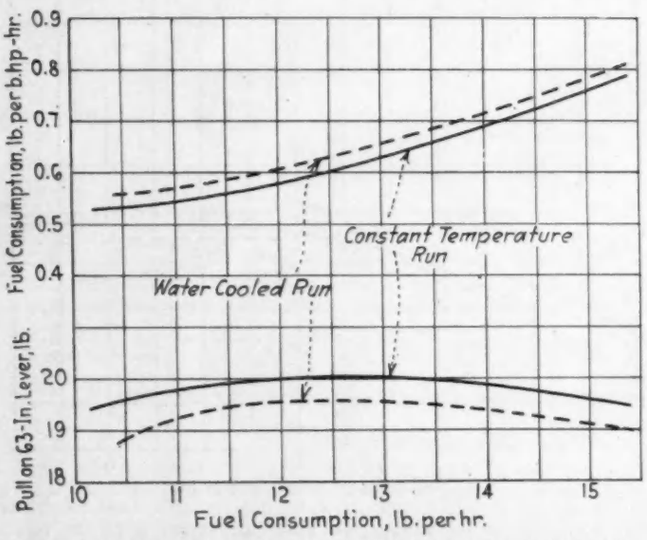


Fig. 8—Relation of Torque to Air-Fuel Ratio at 1000 R.P.M. At This Engine Speed There Is No Difference in the Effects of Constant-Temperature and Water-Cooled Operation

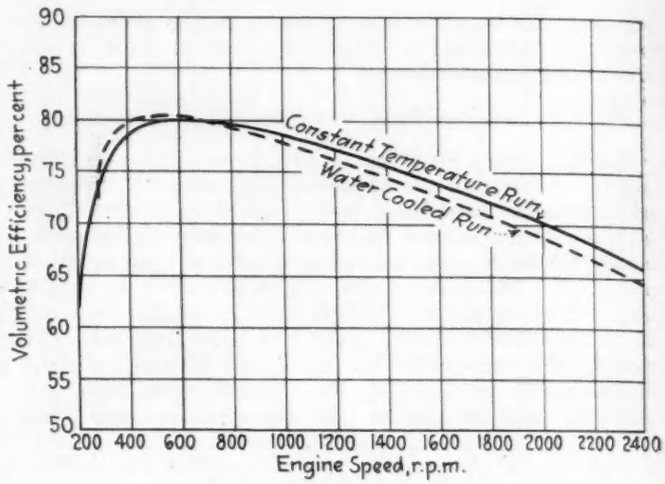


Fig. 9—Volumetric Efficiency with Water-Cooling and Constant Temperature

The Engine Runs Were Made with Water-Cooling at 170 Deg. Fahr. and Constant Temperature at 212 Deg. Fahr. The Difference in Volumetric Efficiency Indicated Is Only 2 Per Cent, Which Is Within the Allowable Error of the Air-Meter and Makes It Apparent That the Higher Average-Temperature Does Not Affect the Volumetric Efficiency of the Engine Materially

212 deg., but drops off rapidly between 212 and 250 deg.

The indications for brake pull apparently bring out the fact that the torque of the water-cooled engine at 170 deg. temperature is equal to the torque with constant-temperature cooling at 225 deg., and that the torque at 135 deg. water-cooling is equal to that of 212 deg. constant temperature.

Further analysis of information given in Fig. 11 shows that, when friction is added to brake pull over the entire temperature-range, some of the differences indicated in brake-pull are accounted for. Thus, the torque advantage for constant temperature at 212 deg. as compared with 170 deg. for water-cooling, is 0.4 lb. at 63 in. The friction advantage of constant temperature at 212 deg. as compared with 170 deg. water-cooling is 0.35 lb. at 63 in. Therefore, the indicated pulls at 170 deg. water-cooling and 212 deg. constant temperature are practically the same. These curves show a marked advantage in indicated pull at lower water-temperature; however, the difference is absorbed by higher friction of the water-cooled engine, which reduces the brake pull, as shown in the curves.

These curves also indicate, apparently, that the water-cooled engine is least sensitive to temperature changes between 160 and 170 deg., while with constant temperature it is least sensitive to temperature changes between 184 and 212 deg. This is the only variation that could be considered for constant temperature, which would be the difference caused by using alcohol as an anti-freeze solution in winter and water in summer.

Water-Level and Steam-Dome Capacity

Inasmuch as the particular cooling-system used in the foregoing test was of the type that includes a dome in the cylinder-head, its governing proportions may as well be examined. Fig. 12 indicates the normal water-level and normal steam-dome of a cold engine and the relative rise of the water-level that occurs as a result of expansion and the volcanic action of the boiling water. With a proper normal water-level, a condition is obtained in which water that is being carried over by the steam from the engine-block to the radiator fills about one-half the outlet pipe, which in our tests

was 1½ in. in diameter, and water flowing from the radiator to the block fills about one-half the inlet pipe, which was also about 1½ in. in diameter.

These conditions, as shown in Fig. 1, are desirable because, with an air space above the water-level in the water-outlet tube, it is possible for the steam to flow into the radiator without surging. If this tube is allowed to fill with water, it becomes necessary for the steam to break through the water in spurts, causing a surging action, which, although not important, is

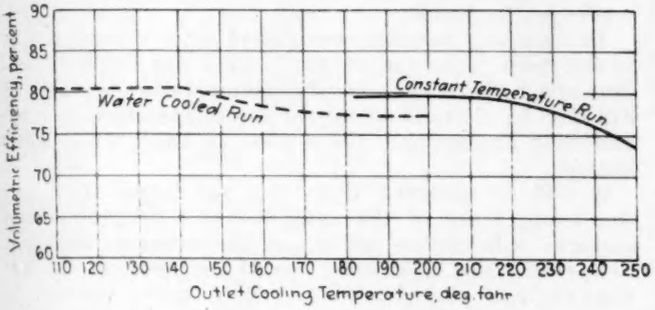


FIG. 10—EFFECTS ON VOLUMETRIC EFFICIENCY OF VARIATIONS IN COOLING TEMPERATURE

With Water-Cooling, the Volumetric Efficiency Is Highest and Constant between 110 and 140 Deg. Fahr. and Drops 2.5 Per Cent between 140 and 180 Deg. With Constant-Temperature Operation, It Is Highest and Constant between 184 and 212 Deg. and Drops 1 Per Cent between 212 and 225 Deg. and 6 Per Cent between 225 and 250 Deg. The Slight Advantage Is with Water-Cooling between 110 and 140 Deg.

unnecessary. It is advisable to use for water only half the capacity of the pipe connecting the radiator with the water-pump to eliminate the possibility of an air-trap in the pump and to permit air to escape between the water and the top of the tube. This condition has been found necessary, as otherwise an intermittent flow is set-up instead of a continuous circulation.

Study of the foregoing conditions for the purpose of determining the proper water-level, which means the percentage of steam-dome capacity to a given quan-

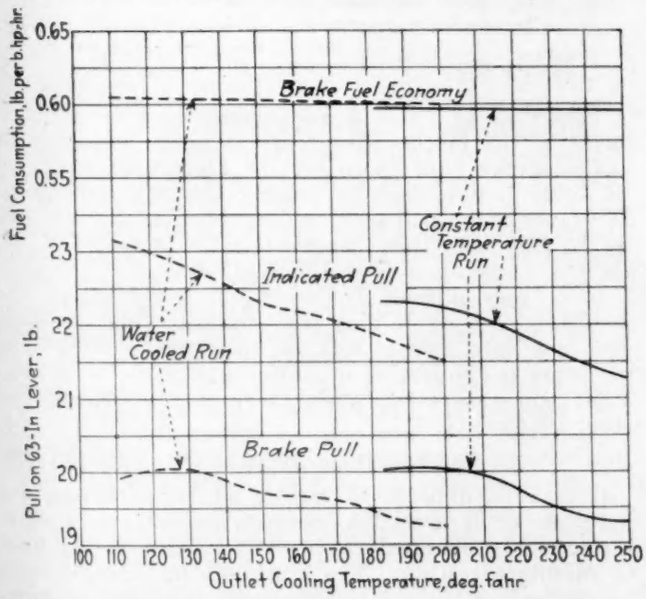


FIG. 11—EFFECT OF TEMPERATURE ON TORQUE AND INDICATED ENGINE-PULL

The Torque with Water-Cooling at 135 and 170 Deg. Fahr. Is Equal to the Torque with Constant Temperature at 212 and 225 Deg., Respectively, But When Friction Is Added the Torque at 170 Deg. Water-Cooling and 212 Deg. Constant Temperature Is Practically the Same

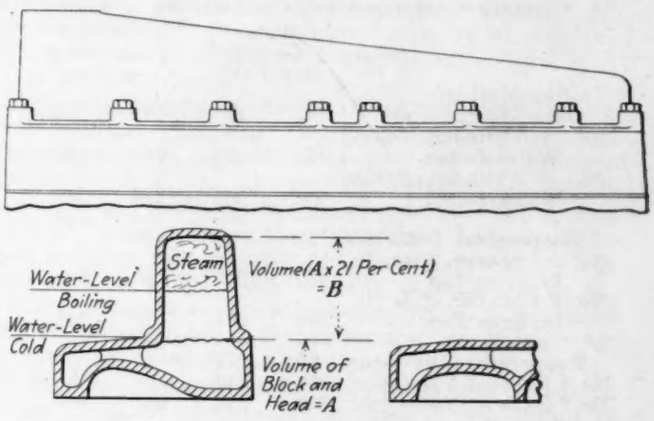


FIG. 12—STEAM DOME OF CONSTANT-TEMPERATURE ENGINE

The Cold-Water Level Is Raised between 12 and 15 Per Cent When the Engine Is Run, Due to Expansion and Volcanic Action of the Water. The Steam-Dome Capacity Should Be Equal to 21 Per Cent of the Water Content of the Engine-Block. It Is Reduced between 6 and 9 Per Cent by Water Expansion

tity of water, brought out the fact that, in the particular engine used, a steam-dome capacity equal to 21 per cent of the normal quantity of water in the block would give the proper proportion in the quantity of water and steam passing into the radiator. If an engine, when filled to the proper level, contains 6 qt. of water, then 21 per cent of 6 qt. should be the equivalent capacity of the steam-dome.

The quantity of water contained within the job proper should be controlled by the minimum core space

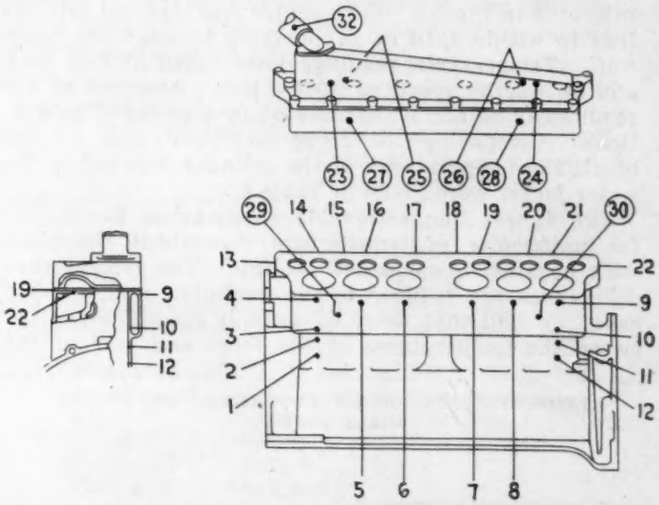


FIG. 13—POINTS WHERE ENGINE-METAL TEMPERATURES WERE TAKEN

Numbers in the Circles Indicate Points at Which the Temperature Was Taken by Thermometer; at All Others It Was Taken with Thermocouples

required by the foundry and needed for good construction around the valves and by a water-level that will cover the combustion-chamber and spark-plug boss. Approximately 6 qt. of water was used in the engines in our tests, with a normal water-level for a cold engine as shown in Fig. 12. However, the smallest possible quantity of water is the right quantity to use, as the quantity controls the time needed to bring the engine to its proper operating-temperature. With the engine operating, the expansion and volcanic action of the water raised the water-level the equivalent of 12 or 15 per cent of the normal water-content. Thus, under operating conditions, the steam-dome capacity is reduced 6 or 9 per cent. It can be readily seen from

TABLE 6—TEMPERATURES OF CYLINDER IRON

	Water Cooled, Deg. Fahr.		Constant Temperature, Deg. Fahr.	
Cooling-Medium Outlet-Temperature	140	170	184	212
No. 1 Cylinder, below Water-Jacket	179	195	213	233
No. 6 Cylinder, below Water-Jacket	192	211	216	233
Temperature Difference	13	16	3	0
No. 1 Cylinder, Side, ½ In. from Top	180	207	226	245
No. 6 Cylinder, Side, ½ In. from Top	200	224	225	248
Temperature Difference	20	17	1	3
No. 1 Exhaust-Valve Seat	226	249	237	250
No. 6 Exhaust-Valve Seat	202	228	239	257
Temperature Difference	24	21	2	7
No. 1 Intake-Valve Seat	195	219	224	244
No. 6 Intake-Valve Seat	177	203	219	237
Temperature Difference	18	16	5	7

Fig. 12 why a great quantity of water must pass into the radiator when the steam-dome capacity is less than that indicated.

Temperatures of Cylinders and Valve-Seats

As a means of bringing out the relative effects of operating-temperatures and cooling systems on the temperature of the actual iron of the cylinder-block, readings were taken with a Leeds & Northrup potentiometer at 22 different locations of thermocouples, as indicated in Fig. 13. Each couple was inserted into the iron to within 1/16 in. of breaking through the inner wall. Temperature readings were taken at full load, and an engine speed of 1000 r.p.m. Analysis of the results can be accomplished best by a series of tabulations. Comparing the front and rear ends of the block, the temperatures of the cylinder iron below the water-jacket were given in Table 6.

The figures unquestionably substantiate the claims for uniformity of temperature throughout the block with constant-temperature cooling. The temperature differences are reduced to the minimum, and in some cases we find that there is actually no difference between the temperatures of the front and rear of the cylinder-block under constant-temperature cooling conditions. The variations as indicated for water-cooling are taken under ideal conditions for this system, as an analysis of water-temperature differences throughout the block under operating conditions will show. For instance, the water temperature, with an outlet temperature of 140 deg. and inlet temperature of 135 deg., shows a total variation of only 6 deg. throughout the block. With an inlet temperature of 164 deg. and an outlet temperature of 170 deg., there is a maximum total temperature-difference of 6 deg. throughout the water in the block.

To meet any possible speculation as to a comparison of the temperatures in the iron with a water-cooled system and those with a constant-temperature system at 212 deg. in districts where the air temperature is high and road grades long, the figures in Table 7 are submitted.

It will be observed from this tabulation that (*a*) the temperature of the cylinder-block becomes more uniform with water-cooling, as the average temperature rises; (*b*) at a water-cooled temperature of 200 deg., the rear end of the block, ½ in. below the top, is as hot as the entire block is under constant-temperature operation at 212 deg.; it is not unusual to run a water-cooled engine at 200 deg., hence operating temperatures as high as those attained with the constant-temperature system are not uncommon; (*c*) the temperatures of the exhaust-valve seats are higher with water-cooling than with constant-temperature operation at 212 deg.; and (*d*) the temperature of the intake-valve seat, insofar as Nos. 1 and 6 cylinders are concerned, decreases in uniformity with water-cooling as the temperature of the cooling medium rises from 190 to 212 deg.

TABLE 7—ENGINE OPERATING-TEMPERATURES IN HOT HILLY DISTRICT

	Water Cooled, Deg. Fahr.		Constant Temperature, Deg. Fahr.	
Cooling-Medium Outlet-Temperature	190	200	212	212
No. 1 Cylinder, below Water-Jacket	210	218	230	233
No. 6 Cylinder, below Water-Jacket	216	224	234	233
Temperature Difference	6	6	4	0
No. 1 Cylinder, ½ In. from Top of Block	221	232	247	245
No. 6 Cylinder, ½ In. from Top of Block	234	245	254	248
Temperature Difference	13	13	7	3
No. 1 Exhaust-Valve Seat	252	257	264	250
No. 6 Exhaust-Valve Seat	240	248	260	257
Temperature Difference	12	9	4	7
No. 1 Intake-Valve Seat	231	242	252	244
No. 6 Intake-Valve Seat	217	225	234	237
Temperature Difference	14	17	18	7

TABLE 8—VARIATION IN UNIFORMITY OF INTAKE-VALVE-SEAT TEMPERATURES WITH INCREASE IN WATER-COOLING TEMPERATURE

	Water-Cooled, Deg. Fahr.				
Cooling-Medium Outlet-Temperatures	140	170	190	200	212
No. 1 Intake-Valve Seat	195	219	231	242	252
No. 6 Intake-Valve Seat	177	203	217	225	234
Temperature Difference	18	16	14	17	18

Referring to Table 6 and combining the temperatures of the intake-valve seats at lower water-cooled temperatures with those for the higher temperatures in Table 7, it is seen by Table 8 that the uniformity increases with increased cooling-medium temperature up to 190 deg., then decreases with increase of the cooling temperature from 190 to 212 deg.

It is very difficult tr record oil temperatures for peratures as a whole that, with water-cooling, we must deal not only with a lack of uniformity but with a variation in the degree of uniformity, while with constant-temperature operation there is practically constant uniformity.

How Water Temperature Affects Oil Temperature

It is very difficult to record oil temperatures for purposes of information, because almost any desired result can be obtained, particularly when fans are used to facilitate testing. The truth of this can be seen by comparing the oil temperatures indicated in runs that are made for other purposes. Some runs show higher oil-temperature for water-cooled operation at 170 deg. than for constant-temperature operation at 212 deg., while others show higher oil-temperature for

constant-temperature runs at 212 deg. than for water-cooled runs at 170 deg. Consequently, observations of oil temperatures were made under conditions in which great care was taken to insure that no change occurred in any external factor that might affect the oil temperature. Table 9 shows the results. The indication from this is that the temperature of the oil is not affected by the system of cooling but by the temperature of the cooling medium.

TABLE 9—COMPARISON OF OIL TEMPERATURES

Outlet Water Temperature, Deg. Fahr.	Oil Temperature, Water-Cooled Deg. Fahr.	Oil Temperature, Constant Temperature, Deg. Fahr.
110	136	
130	140	
140	140	
150	140	
160	140	
170	145	
180	148	149
190	150	
200	152	
212	153	153

Large Water-Flow Aids Temperature Uniformity

When constant-temperature cooling is accepted, there will be a difference of opinion as to what type of condenser or what specific system is most desirable. The main difference of opinion will, I believe, be on the question whether or not water should be passed into the condenser or radiator with the steam that is vaporized in the engine-block, and, if so, how much. This question arose some months after the tests whose results have been given in the foregoing part of the paper were made. Therefore another set-up was made in which the engine could be operated at constant temperature or with water-cooling and the quantity of water circulated could be controlled independently of the engine speed.

This test was made to determine the effect on the temperature of the metal of the circulation of various quantities of water, and readings were taken of the metal temperature by thermocouples and also of the pounds of water circulated per brake-horsepower per minute. When operating on the constant-temperature system the registration holes between the cylinder and the head were readjusted to suit constant-temperature operation; when operating on the water-cooling system the registration was used as found. The results are given in Table 10.

These figures indicate that local stagnation occurs to some extent with an insufficient flow of water in the steam-cooling or constant-temperature system, notwithstanding the general communication of the water holes from the block to the head. The average temperature of Nos. 1 and 6 exhaust-valve seats with 2 lb. of water circulated per b.hp. per min. is 396 deg. For 3 lb. per b.hp. per min. the average temperature is 344 deg., for 4 lb. per b.hp. per min. it is 352 deg., and for 5 lb. per b.hp. per min. it is 350 deg.

It appears that when the flow was increased from 2 to 3 lb., the greatest improvement in temperature uniformity was made and, incidentally, that the lowest-average temperature was reached. Uniformity seems to increase with increase in the flow of water in the constant-temperature system, despite the fact that the average temperature appears to be from 6 to 8 deg. higher. The data indicate a pump capacity of approximately 150 lb. per min. for a 40-hp. engine, and this capacity should be borne in mind when considering pump design.

The temperatures in Table 10 include readings for water-cooling at 180 deg. Temperature uniformity for this condition is certainly very poor for the particular engine used, but, as previously stated, the engine was run on the water-cooling-system just as it was built. It is to be noted that the temperature uniformity is not improved by increased water circulation. The difference in temperature actually increased with increased water-flow. In all probability, the increased volume of water upset the calibration or at least aggravated an existing bad condition.

When the maximum quantity of water was circulated in the constant-temperature system, it was found to aid uniformity of temperature between the front and rear ends of the block. This can be explained by recalling that, for constant-temperature operation, the water registration holes between the cylinder-block and cylinder-head were all over the block and head so as to permit the ready use of steam, which evidently is carried upward by the water. This means a greatly increased total-passage area from the block to the head, which possibly minimizes the effect of an overabundance of water. On the other hand, in the water-cooling-system, as with all water-cooled engines, the calibration of the holes is based upon the greatest concentration of flow at the hottest point. This seems to restrict the total-passage area.

The figures serve to support the contention that water-flow is essential to ideal operation, and that the original conception that 100 per cent of steam in the condenser is desirable was erroneous.

TABLE 10—EFFECT OF WATER-FLOW ON TEMPERATURE ENGINE SPEED, 1600 R.P.M.

	Constant-Temperature Operation at 212 Deg.			
Water Circulated per minute per Brake Horsepower, lb.	2	3	4	5
	Temperature Readings, Deg. Fahr.			
No. 1 Exhaust-Valve Seat	445	330	346	348
No. 6 Exhaust-Valve Seat	347	358	357	352
Difference	98	28	11	4
	Water-Cooled Operation at 180 Deg.			
Water Circulated per minute per Brake Horsepower, lb.	2			5
	Temperature Readings, Deg. Fahr.			
No. 1 Exhaust-Valve Seat	297			281
No. 6 Exhaust-Valve Seat	353			356
Difference	56			75

PART II BY L. P. SAUNDERS[2]

Car operation under constant temperature is similar, in general, with all species of this type of cooling except when, with extremely experimental outfits, considerable water is lost or when a vacuum is maintained in the system to lower the temperature of the cooling medium. With reference to the latter, we never have believed that the end was worth the means, hence will not refer to it again. It is assumed that the industry never will adopt a cooling system that permits constant loss of water, and, as it is easy to retain the

[2] M.S.A.E.—Experimental engineer, Harrison Radiator Corporation, Detroit.

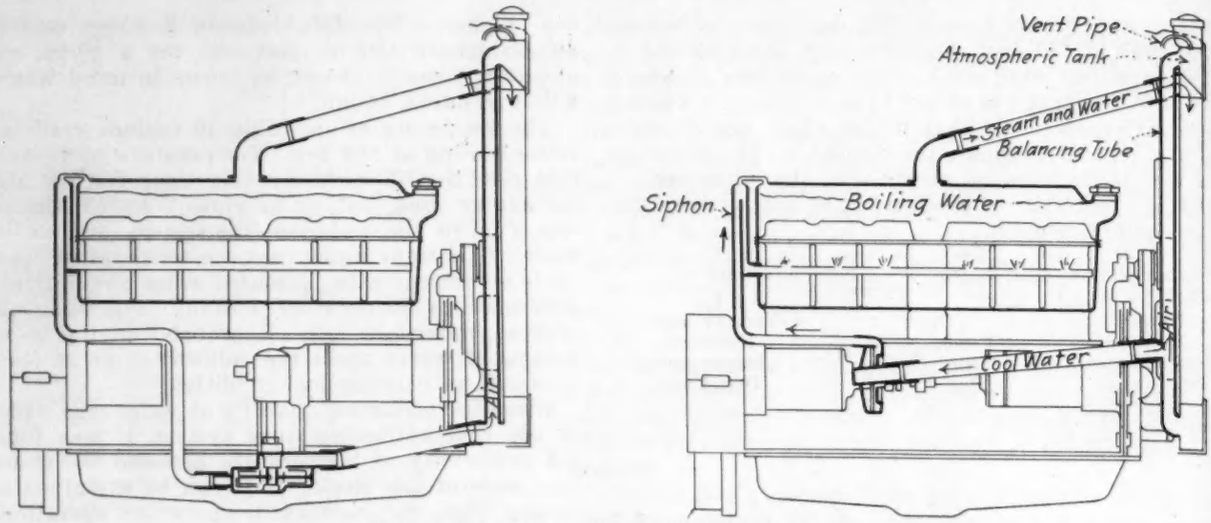


FIG. 14—WATER-PUMP VENTED TO AVOID AIR-TRAP

The Pump Is So Located and the Connecting Pipe to the Radiator Is of Such Size That the Pipe Is Only Half Filled with Water, Hence Air Can Escape above the Water. In the Drawing at the Right Is a Detail Showing a Balancing Tube on the Condenser for Use when the Location of the Pump Does Not Allow the Condensate To Flow to the Pump

water, discussion will be confined to road-tests of car operation with a constant-temperature system that includes a satisfactory condenser.

Evidence is not lacking at this time of the year that present cars are over-cooled in winter. Radiator covers are in use in a wide variety of form, from improvised cardboard shields to decorative shutters upon which a commercial business has been built. Engineers have long recognized the necessity of preventing over-cooling and the difficulties presented by the problem of maintaining a suitable operating-temperature under widely varying climatic and barometric conditions and under light and heavy engine-loads.

Observed winter temperatures of the outlet water from the engine-block of a water-cooled engine, with a radiator large enough to cool the engine satisfactorily when the car is driven hard in high altitudes, were: 100 deg. fahr. at 35 m.p.h. at 25 deg. fahr. atmospheric temperature, 132 deg. fahr. when running with wide-open throttle at 20 m.p.h. and air temperature of 25 deg. and 150 deg. fahr. at 15 m.p.h. in second speed with wide-open throttle and 25 deg. air temperature. At a water temperature of 150 deg., alcohol used as an anti-freeze solution is evaporated quickly.

Controlling the water circulation by a thermostat has disadvantages. The liability of the water to freeze is increased by impeding its circulation, and the temperature for which the thermostat must be set is too low for satisfactory engine operation unless it is set so high that alcohol must be lost.

The normal demands of automobile operation cover a very wide range, as operation is affected by density and temperature of the atmosphere and by load on the engine, hence satisfactory improvement in operation will be hampered unless means are provided for permitting efficient operation over the necessarily wide range. The Muir system of constant-temperature cooling which has been developed for this purpose permits successful operation under all conditions of atmosphere, altitude and load.

DIFFICULTIES MET AND OVERCOME

The chief difficulty encountered in developing the system was the prevention of water loss, and the cross-flow type of condenser was developed to avoid this. To realize how much water passes over with the steam and the necessity of giving it right of way, one needs only to disconnect the steam inlet to the radiator during the operation of a constant-temperature engine. It is, therefore, obvious that the greatest possible length of cooling tubing is necessary. The conventional radiator contains just about sufficient. The cross-flow condenser permits the use of practically the full length of condensing tubing without danger of the steam forcing the water that is ahead of it out into the atmosphere. A number of water-cooled cars on the market are equipped with radiators whose top tanks are designed so that the loss of comparatively little water uncovers some of the tube ends and admits air, which displaces more water until finally the pump becomes air-bound and circulation of the water ceases. In some other systems the tanks are constructed so that the water-flow reaches a critical velocity and a vortex is formed at the tube ends, so that air is admitted and water is forced out.

A constant-temperature system is not so sensitive to water loss as the water-cooled system. In fact, loss of water may continue until the water-level falls below the combustion chamber and "pinging" and loss of engine power result. A car fitted with the constant-temperature system and having the normal water-content of 3.3 gal., starting at the bottom of an 11.6-per cent grade at 10 m.p.h., stalled after ascending 492 ft. When the quantity of water was reduced to 1 gal., the car stalled at 80 ft. On a run of 17 miles around a track at 35 m.p.h., the power loss increased with the mileage, the engine began to miss rather badly and was stopped. The water content was then found to be 3 qt.

Considerable difficulty was encountered in venting the down-flow type of radiator. When the car was operated with comparatively dry steam, the condensation was not sufficient, under ordinary driving conditions, to keep the pump primed and it became air-locked. As such an air-lock will force all the water out of the cooling system, provision should be made in the design of the condenser against any possibility of an air-lock occurring. If a sufficient quantity of water was driven over with the steam, the pump operated satisfactorily so long as the speed of the pump sufficed to exhaust the lower tank enough to uncover

TABLE 11—COMPARATIVE WATER-TEMPERATURES IN CROSS-FLOW AND STANDARD RADIATOR-CORES, BOTH USED AS WATER-COOLERS

	Car A			Car B	
Car Speed, m.p.h.	20	35	45	20	35
	Water Temperatures, Deg. Fahr.				
Standard Core	193	155	161	153	160
Cross-Flow Core	188	152	153	143	149
Difference	—5	—3	—8	—10	—11

the lower ends of the tubes and the vent, but when the lower ends of the tubes and vent were covered, the water was lost.

When the pump was properly vented, a balance would be struck between the pump capacity and the amount of water passing over; the system would regulate itself. However, the problem was presented of convincing the industry that, when a loss of water was indicated during the period of regulation, the rejection of water would cease when the true balance was reached. Venting the pump is accomplished by having the connecting pipe between the condenser and the pump of such size and so located that the pipe is filled with water to only one-half its capacity and air will be permitted to escape above the water, as shown in Fig. 14. In the drawing at the right is shown a condenser provided with a balancing tube for use when the location of the pump does not permit the condensate to flow to the pump. This condition requires a pressure of from ¼ to ½ lb.

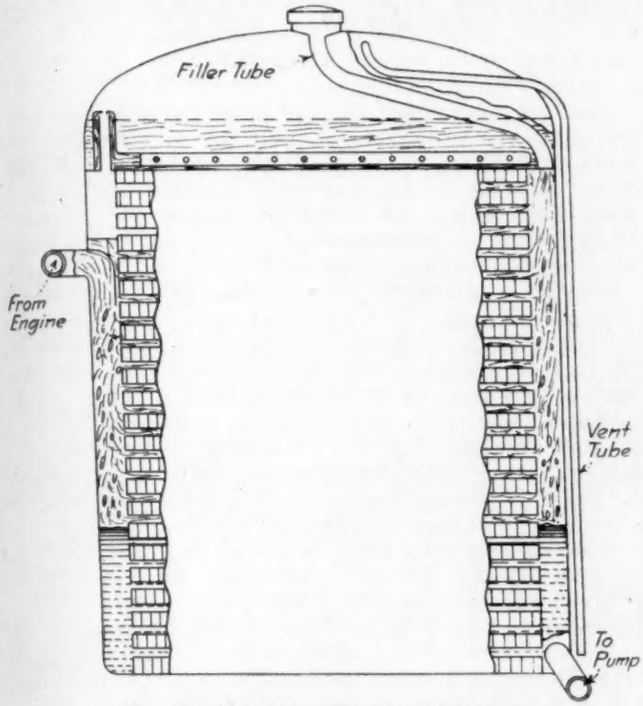


FIG. 15—CROSS-FLOW TYPE OF CONDENSER
The Entire Cold Side Is Vented and Remains So Irrespective of the Water-Level. Road-Tests Show That, When Completely Filled, It Will Operate Satisfactorily as a Water-Cooler

TABLE 12—TEMPERATURES IN CROSS-FLOW CORE USED AS A WATER-COOLER AND AS A CONSTANT-TEMPERATURE SET-UP

Car Speed, m.p.h.	20	35
	Water Temperatures, Deg. Fahr.	
Water-Cooled	143	149
Constant Temperature	100	155
Difference	—43	+6

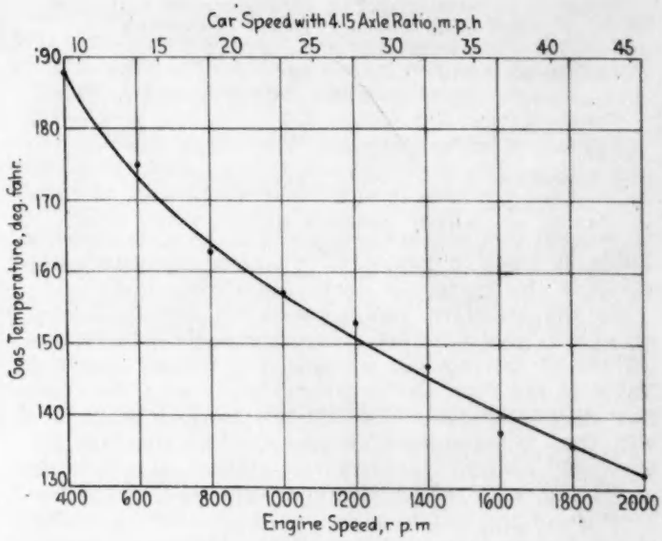


FIG. 16—EFFECT OF STEAM-JACKETED MANIFOLD ON FUEL-MIXTURE TEMPERATURE
Jacketing the Intake-Manifold Gives a High Gas-Temperature at Light Engine-Load and Low Gas Temperature at High Engine-Load

The cross-flow condenser shown in Fig. 15 has none of the foregoing drawbacks, since the entire cold side is vented and remains so irrespective of the water-level. Road-tests with this type of condenser brought out the fact that, when completely filled, it will operate satisfactorily as a water-cooler; in fact, it possesses the additional advantage of having a cold overflow, the water being forced across the core by the pump without the agitation that usually occurs in the top tank of the radiator of a water cooling-system.

The temperature figures in Table 11 will show how this cross-flow core compares as a water-cooler with a standard water-radiator on two different makes of car. The core was the same size for each set-up, and all temperatures are corrected to 75 deg. fahr. air temperature.

A comparison of the temperatures in the cross-flow core when used as a water-cooler and when used as

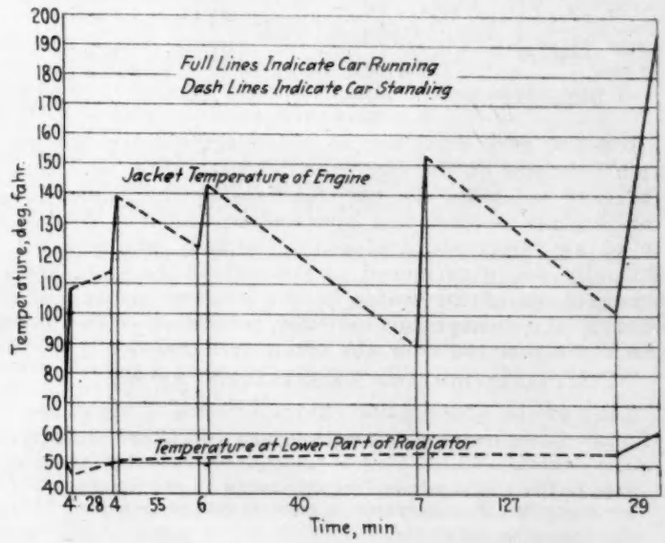


FIG. 17—RISE AND FALL OF WATER TEMPERATURE DUE TO STARTING AND STANDING
The Temperatures Were Taken in Winter with a Car Fitted with a Constant-Temperature Radiator-Core. They Show a Range, during 6 Hr. and 40 Min., of 70 to 195 Deg. Jacket Temperature and 46 to 56 Deg. in the Lower Part of the Radiator

TABLE 13—TEMPERATURES IN STANDARD CORE AND CROSS-FLOW CORE UNDER EXTREME CONDITIONS OF DRIVING AND ALTITUDE

Car Speed, m.p.h.	20	35
	Water Temperatures, Deg. Fahr.	
Standard Core	153	160
Cross-Flow Core	100	155
Difference	—53	—5

a constant-temperature set-up, with the water-level reduced, is made in Table 12. These temperatures were taken at the vented, or cold, side of the core.

As the standard radiator-core on car *B* is large enough to give satisfactory cooling under extreme conditions of driving and altitude, the results shown in Table 13 are obtained by comparing it with the cross-flow core used as a constant-temperature set-up. It will thus be seen that the constant-temperature system will operate satisfactorily under all conditions of driving, with the additional advantage that the temperature of the water in the engine will be maintained at 212 deg. so long as water alone is used as a cooling medium. This temperature will be reduced only when alcohol is used as an anti-freeze solution or the boiling point of the water is lowered by increase in altitude and consequent reduction in atmospheric pressure.

How Increased Heat Affects Carburetion

The comparison in Table 14 of fuel consumption as affected by the difference in temperature of the cooling medium when operating with the standard core, the cross-flow core used as a water-cooler and the

TABLE 14—COMPARISON OF FUEL CONSUMPTION

Car Speed, m.p.h.	25	35
	Fuel Consumption, Miles Per Gal.	
Standard Core, Water Cooling	20.11	18.86
Cross-Flow Core, Water Cooling	20.11	19.01
Difference	0.00	0.15
Cross-Flow Core, Water Cooling	20.11	19.01
Cross-Flow Core, Constant Temperature	21.36	19.98
Difference	1.25	0.97
Standard Core, Water Cooling	20.11	18.86
Cross-Flow Core, Constant Temperature	21.36	19.98
Difference	1.25	1.12

cross-flow core used for constant-temperature operation indicates an increase in miles per gallon with the increase in heat. A car fitted with the constant-temperature system was operated in Imperial Valley in an air temperature of about 100 deg. fahr. and no difficulty was experienced as a result of the increased temperature of the water in the cooling system, although the temperature of the condensed water in the down-flow radiator was often over 200 deg.

Acceleration and Deceleration Effects

Tests of the acceleration characteristics of a number of cars have demonstrated that any slight advantage

TABLE 15—COMPARATIVE DECELERATION WITH WATER-COOLING AND CONSTANT-TEMPERATURE OPERATION

Car Speed, m.p.h.	25	35	45
	Deceleration Time, Sec.		
Standard Core, Water Cooling	31.0	38.0	49.0
Constant Temperature	31.7	42.7	52.5
Difference	0.7	4.7	3.5

is in favor of constant-temperature operation as compared with water-cooled operation. Deceleration tests of an eight-cylinder car during the winter gave the results in Table 15, in which the figures indicate the time in seconds required to decelerate to 5 m.p.h. from the speeds shown, and are averages of six runs made in each direction on the straightaway of the General Motors proving grounds. The differences check with results obtained on the dynamometer, in which the indicated power is lower with the constant-temperature system as compared with water-cooling.

TABLE 16—COMPARISON OF FUEL CONSUMPTION BY AN ENGINE WITH EXHAUST-HEATED MANIFOLD

Car Speed, m.p.h.	10	20	30	40
	Fuel Consumption, Miles Per Gal.			
Water Cooling	10.6	9.6	9.7	8.8
Constant Temperature	18.1	15.2	13.8	12.1
Difference	7.5	5.6	4.1	3.3

This machine was equipped with a carbureter on the left side of the engine and the manifold to the various ports on the right side passed through the water-jacket in the head, with an exhaust hot-spot just above the carbureter. The disadvantage of water-jacketing the manifold, as shown by the figures in Table 16, is that more time is required to warm-up than with a hot-spot, although it gives the desirable condition of high gas-temperature at light load and low temperature at high load, as indicated in Fig. 16. The figures in Table 16 were taken in each case with a carbureter-setting that gave an acceleration from 5 to 25 m.p.h. in 9.6 sec. and are averages of six runs in each direction on the straightaway. The large differences are due entirely to poor distribution, for the results given in Table 14 were obtained from a machine that had a very efficient manifold-heating system.

Hill-climbing tests were made with the eight-cylinder machine on the 7.26-per cent grade at the proving grounds, and the averages of six runs, as given in Table 17, show that the constant-temperature cooling system is slightly better than water-cooling. The difference in performance is attributed to the better distribution obtained with the cross-flow system, for, when an at-

TABLE 17—COMPARISON OF HILL-CLIMBING PERFORMANCE

	Water Cooled	Constant Temperature	Difference
Starting Speed, m.p.h.	10.00	9.75	—0.25
Maximum Speed, m.p.h.	28.50	32.00	+3.50
Speed at Top of Grade, m.p.h.	21.00	26.00	+5.00
Time of Run, sec.	69.00	64.30	—4.70

tempt was made to climb the 11.6-per cent grade, it became necessary, in order to obtain consistent results, to drive down the 7.26-per cent hill and along the main track before starting up the 11.6-per cent grade at a speed of 10 m.p.h. The up-grade distance traveled then in each of six runs before the car stopped was 1300 ft. It was necessary to drive the car in this way because the radiator was cold, and the cold air passing through it chilled the aluminum manifold so much that the carbureter would "spit." When the car was run as a water-cooled machine, the stalling distance varied from 400 to 900 ft. and, although a number of runs were made, the car did not stall twice at the same place. The water temperature in these tests was approximately 130 deg. fahr.

TABLE 18—RADIATOR TEMPERATURES AT VARIOUS CAR-SPEEDS

Car Speed, m.p.h.	15	20	35	50
	Temperature of Radiator Water, Deg. Fahr.			
Standard Core, Water Cooled	221	195	159	172
Cross-Flow Core	175	188	159	165
Difference	—46	—7	0	—7

Increased Hood-Ventilation Desirable

In some automobiles the temperature of the air in the driving section of the body becomes uncomfortably high and, although present ventilators are adequate with water-cooling, it may be necessary, when the temperature of the whole mass of the engine is raised, to provide additional ventilation or insert deflectors to prevent the heated air from passing through the pedal slots and other openings into the body of the car. The provision of ample outlet area for the air to pass from under the hood will mitigate this trouble and also reduce the temperature of the air that enters the carbureter, which should increase the volumetric efficiency.

It is not believed that the size of the radiator-core can be reduced, because the coolers on a majority of cars are inadequate to meet the requirements of extreme conditions of driving and of air temperature and density. The figures in Table 18 show that the standard core is not large enough for severe driving conditions.

TABLE 19—RADIATOR TEMPERATURES AT VARIOUS CAR-SPEEDS WITH FANS OF DIFFERENT SIZE AND THE SAME RADIATOR-CORE

Car Speed, m.p.h.	15	20	35	50
	Radiator Temperature, Deg. Fahr.			
16-In. Fan	218	187	153	166
17½-In. Fan	197	179	151	159
Difference	—21	—8	—2	—7

The size of the fan will remain at least as large for constant-temperature cooling as at present for water-cooling. The difference in results obtained with fans of different size, used with the same core, are given in Table 19.

The effects of starting and standing in cold weather upon a car fitted with a constant-temperature core are shown in Fig. 17, in which the rise of temperature of the jacket-water during starting periods is shown in the upper solid lines, and the drops in temperature during the standing periods, subsequent to the initial warming-up period, are shown in broken lines. Corresponding changes in temperature of the water in the lower part of the radiator are indicated in a similar manner at the lower part of the chart.

Rapid Water-Flow Improves Hill-Climbing

To ascertain the correctness of a belief that the volume of water circulated in the system per brake-horsepower per minute seriously affected the performance of the engines, two cars were provided with different water-pump capacities, one with a delivery of 1.7 lb. per b.hp. per min. and the other with a delivery of 3.4 lb. per b.hp. per min. The original Muir installation had both a circulating and a make-up pump; the present design permits the single pump to perform both duties.

Road-tests of these two machines on a hill with an average grade of 9.2 per cent demonstrated that the first machine had a maximum speed of 22 m.p.h. and a very intense spark ping, and, at the end of the climb, the engine overran for more than 1 min. after ignition was switched off and with the throttle closed. The second car had a maximum speed of 26 m.p.h., the intensity of the spark ping was very much reduced as compared with that of the first car and the engine did not overrun upon completion of the climb. Both runs were made under identical conditions, with the same driver and two passengers.

THE NATURAL FLOW OF CAPITAL

CONSIDERING the relative supplies of capital in Europe and here, it would not be a natural or normal movement to have capital transferred from any of the debtor countries on the continent of Europe to this Country at this time. Liquid capital is desperately scarce throughout most of Europe, as current rates show. The very act of transferring capital from there to here under present conditions would tend to create a greater scarcity there and a plethora here, which would cause capital to flow back as certainly as water flows downhill.

The position of the United States in the world is illustrated by that of a rich man whose income is far in excess of his own personal expenditures and who is constantly investing the surplus in new business enterprises. In that manner the surplus is returned to business uses and the equilibrium is maintained. That is the law of economic distribution. The United States is the rich man of the world. Its income is so large and its own development so far advanced that our people are scouring the world to find investments where the needs are greater and the returns are larger. We are leaking capital at every seam.

The economic law and the moral law are one and the same. There is only one law in the universe and that is the law of right adjustment which promotes general social progress. I think it all to the good that neither individuals nor nations can indefinitely pile up wealth strictly for themselves, without regard to the needs of others.

I do not think that our foreign loans and investments will ever be liquidated in the sense that we will actually collect and withdraw within our boundaries the capital that we have abroad. I do not suppose the Pennsylvania Railroad will ever pay off its bonded debt completely and fully. The foreign investments of the United States probably will be as large, perhaps larger, 50 years from now than they are today, because we will be a richer people and still looking for investments. Possibly in time our foreign investments may be shifted to other countries, more in need of capital.

I think that we must adjust our minds to regard the world as our economic field of enterprise, and the economic organization of the world as one organization, and endeavor to improve upon the political engineering talent that has been managing it.

There is no limit to the amount of work to be done in the world, or to the amount of business to be had, or to the amount of wealth that can be created from the resources of nature. The purchasing power of every country is in its own powers of production, and the greatest prosperity for every country is to be had in connection with the prosperity of all other countries.

There can be no such thing as general over-production so long as human wants are unsatisfied. What looks like it, is simply unbalanced production.

The great problem of the world is so to organize and integrate and coordinate the resources and industries of the world as to secure the greatest possible production and distribution of all the things that minister to the comfort and welfare of the population. That is the great appeal to the enlightened and constructive people of the world.—From an address by George E. Roberts, before Society of Engineers of Western Pennsylvania.

Design Standardization for Simplified Service

By B. C. Hinckley[1]

Service Engineering Meeting Paper

ABSTRACT

SERVICE became of paramount importance as automobiles became more numerous, and the service departments impressed upon the engineers the absolute need of designing the car so that all unit parts would be accessible and removable without taking down other parts. Standardization and mass production of parts have greatly simplified and cheapened service by making parts for cars of the same make or model interchangeable and by reducing the number of sizes of bolts, nuts and connections so that fewer tools are required for removing and setting them up. Much remains to be done, however, by automotive engineers toward simplifying construction with a view to the tool equipment of the service-department and repair-shops.

The lightening holes in crankshaft gears could be made of standard size and location so that one instead of about 48 types and sizes of gear remover would suffice. It should not be necessary to remove rear axles from beneath the car or move the engine forward in the frame to take out the transmission box, nor to take the engine from the chassis to get at the main engine-bearings. In some cars it is not even necessary now to remove the cylinder-block to do a first-class job of fitting oversize pistons with the aid of portable cylinder-boring machines and cylinder hones. Axle housings might be made in a way that the differential-gears could be aligned without tearing down the entire axle-assembly. Valve-springs should be made accessible for removal of the valves for grinding without taking off the intake or exhaust-manifold, and the valve guides could be better placed. Bearings and bushings in the different unit assemblies are of too great a multiplicity of size, notwithstanding the fact that the bearing manufacturers have done considerable in the way of standardizing the sizes they carry in stock. Rear-wheel hub-diameters and threads should not be changed when a new car model is brought out, as such a procedure necessitates an additional size of hub puller in the service-station, increasing the investment in equipment not only because of the addition but because the price of the special puller must be higher.

General adoption of the flat-rate system for service is forcing consideration of accessibility and time-saving in servicing automobiles and compelling the tool manufacturer to provide special tools that will make repair work easy to perform and save time in the shop. Quickness and ease of servicing and availability of replacement parts are furnishing one of the best selling arguments for cars at the present time.

Manufacturers of special tools are working in constant contact with service-departments and producing tools that are needed for service on each make of car because of the many and diverse makes of engine, clutch, transmission, rear axle, and other assemblies that enter into the complete car. Similar close cooperation of the automotive engineer with the service engineer is needed further to standardize and simplify the parts and their assembly in the vehicle with a view to reducing the number of tools required and facilitating repair work.

[1] Hinckley-Myers Co., Jackson, Mich.

SERVICE that would have to be performed on a car after it had been in use for some time did not seem to be given any consideration by automotive engineers in the early days of automobile building. The engine and its attachments were set in the frame and the rest of the car was built around the powerplant in such a way that, when it became necessary to service the engine, about half of the front end of the car had to be removed before a mechanic could get at a part to remove and repair it. This, of course, involved the expenditure of considerable time and labor that had to be charged to the owner and, as a consequence, the repair bills were very heavy. This retarded the sale of automobiles to a great extent, because the average man could not afford to pay for their maintenance. All of the efforts of the engineers appeared to be expended in providing a product that would run and withstand the tremendous stresses that were put upon it by inexperienced drivers and the roads over which it traveled.

The difficulty of making repairs and replacements became of paramount importance to the service-departments as the building of automobiles progressed, and they impressed upon the engineers the absolute need of designing the car so that all of the unit parts would be as accessible as possible. Only within the last 8 or 9 years has a really concerted effort been made by engineers to design engines and other car parts with a view to low service and maintenance costs. Prior to that time, only two or three large companies had standardized production to such an extent that an automobile could be torn down to the frame and rebuilt into a smooth-running car from parts that were running originally in an entirely different assembly. European builders have not yet standardized, to the extent that Americans have.

How Service Can Be Simplified

Considerable remains to be accomplished in designing automobiles to a greater degree of interchangeability of parts. Approximately four dozen different types of crankshaft-gear remover are on the market. It would seem that the gears could be made with lightening holes of a standardized size and location so that the crankshaft gear could be removed from any of many makes of car with a single tool. Again, to remove the transmission box from many cars still offered in the market, it is necessary to remove the rear axle from under the car or take off the radiator, loosen the engine and draw it forward and possibly also loosen the steering-column to provide room for the engine to slide forward.

A great reform has been brought about in making the engine bearings accessible, and very few makes of American car are now offered for sale that cannot be serviced completely without removing the engine from the frame. A first-class job of fitting oversize pistons can be done now with any of several makes of portable cylinder-boring machine and cylinder hone without removing even the engine block.

Axle accessibility presents a problem on which the engineer might well work with the service man. Aligning the pinion and bevel-gears of the differential in the majority of axles is a matter of touch and guesswork, and with some axles it is impossible to make adjustments after closing the differential and axle housing without tearing down the entire assembly again if a slight hum or other evidence of misalignment develops.

Valves and Guides Too Inaccessible

The practice followed in placing valve-guides is another respect in which the engineer could advantageously work more closely with the service-department. When these guides become worn a leakage of gas occurs and an uneven running of the engine that is annoying results. These faults can be overcome only by putting in new guides or reaming the old guides oversize and fitting oversize valve-stems. Those who have done a job of this kind realize how much of an undertaking it is.

Considerable can be accomplished by making valves more accessible for adjustment and regrinding. Valves often are neglected by even experienced car-owners because they are too inaccessible for anyone but an expert to do any work on them. It is almost impossible to get at the valve-springs on certain L-head engines with detachable heads, when it becomes necessary to regrind the valves, without removing the intake and exhaust-manifold assembly and half a dozen other engine accessories.

Sizes and makes of bearings and bushings used in the different assemblies in the car could well be standardized. Few cars are made with interchangeable ball and roller bearings, although the bearing manufacturers have done much toward standardizing the sizes that they carry in stock. The car builders seem to leave this matter entirely to the individual preference of the engineer. Bushings are of a multiplicity of diameters, lengths and thicknesses. One popular car requires half a dozen or more special reamers in special micrometer dimensions and several reamers in fractional-inch sizes to service the bushings, while all of the bushings on another popular car can be serviced with six different sizes of reamer of standard fractional dimensions. Scarcely three sizes of reamers in the 30-odd sizes that are carried in stock at the present time by the company which I represent are common to more than one make of car.

Engineers Should Work with Service-Men

Greater standardization of parts could be effected in any number of instances that have come to our attention. This is a problem to be worked out by the car builder, with the car engineers working in close cooperation with the service engineers. The problem in construction that confronts the automotive engineer now is to decide which of the following policies is preferable:

(1) Construction that has long life but relatively high production cost

(2) Sacrifice of much possible strength of construction or length of life for low cost of production or of service, or both

(3) Construction that makes it possible, by use of special tools, to maintain low production and service costs with little, if any, sacrifice of structural strength or length of life

Special tools have the widest application, of course, when the tendency of a company is to follow the third policy, the least application in the second policy and an intermediate range in the first policy. In many places a compromise must be made and in many others improvements could be made if the engineer would bear in mind the convenience and cost of service and the requirements for service tools. I have in mind the diameters and threads of rear-wheel hubs. For no vital reason, a new model will have a hub of different diameter and thread than previous models, and this requires the service-station to have a number of different wheel-pullers. This increases the investment that must be made by the service-station, both because more tools must be provided and because the tool manufacturer's prices must be higher. The standardization of bolts and nuts and many other parts through the efforts of the Society has simplified tremendously the tool equipment required by the shops.

One make of car is on the market for which the boast is made by the builder's service-department that every bolt and nut can be removed with a wrench set consisting of four sockets. This is going a long way toward standardization and the elimination of unnecessary tools. Other automobile builders can accomplish much in this same direction.

Flat-Rate System Forcing a Change

The old idea that the engine must be removed from the frame and completely dismantled when it is necessary to change the flywheel or take up the rear crankshaft-bearing cannot prevail much longer. The flat-rate system of charging for service and the flat-price system of paying mechanics by the job cannot be successful when the car design makes so much extra work necessary. Consequently, the almost universal adoption of the flat-rate system has supplied the force to compel consideration of accessibility and of ease of servicing. This obliges the tool manufacturer to design special tools so that the greatest amount of time can be saved by their use in the repair-shop and so that they will make the work easy to perform.

One of the best selling arguments that the car builders ever had was produced by the efforts put forth by the engineers to standardize. The old habit of placing primary emphasis on sales and secondary emphasis on service has been reversed. An illustration of this happened to a friend of mine who made an automobile trip east through New York State and Massachusetts several years ago and on his way back broke a ring-gear in the rear axle of his car. He and his party were delayed for 3 days a few miles east of Buffalo until a replacement ring could be obtained from the factory and installed. I have heard this man remark since then that when buying a new car the first question he asks is, "Can I get service on this car, and where, if anything goes wrong or if it becomes necessary to replace some part?" Obviously, it is very necessary that automotive engineers design the component parts of the car so that each part is as accessible as possible and that replacement of units can be made with proper tool equipment at the lowest possible cost to the car-owner.

Special Tools Help the Service-Station

All car builders realize the advantage of special tools for servicing their own makes of car, and manufacturers of this class of equipment feel considerable satisfaction because the automotive engineer has been able, with the assistance of the service engineer, to overcome the difficulties that were paramount with the older types of car. The work of the special-tool manufacturer brings him into constant contact with the service-departments and he works in close conjunction with the service engineers. He finds that in many instances changes in

design have been made, as a result of representations by the service engineer, so that the car can be serviced properly and cheaply.

Tool equipment must, of necessity, be special for the various makes of car because of the sundry and diverse makes of engine, clutch, transmission, rear-axle, and other assemblies and units that enter into the complete vehicle. The tool manufacturer has attempted, by close cooperation with the service departments of the different car builders, to produce tools that are adequate for all of the various necessary operations in assembling or taking-down the entire car, and by so doing has enabled the car dealer to operate on the flat-rate service system at a profit and at the same time to satisfy his customers by doing repair work at moderate prices.

Engineers Design for Low Production-Cost

Other considerations than facility of service must be taken into account in car designing. The service-man and the tool manufacturer think that this or that part could be changed so that it would be more accessible but, upon investigation, find that the automotive engineers have designed the part for facilitating production or assembly and, although the design may necessitate a special tool for service, it enables the car builder to reduce the price of the car to a figure that is almost beyond comprehension as compared with the experimental model that is turned out in a single unit. Therefore, we should not always criticize the engineers for sundry parts about a car that cannot be worked upon handily, since their problem is to produce a car in which the buyer receives the greatest value for his money.

Service-stations long ago discarded old notions, and with them went old tools and fixtures. The factor that has helped most to bring this change about is the improvement in engine and car design. The engineers have come to realize that their concentrated efforts must now be made with broad consideration of the work of the repairman and of his tools. Each year the servicing of cars is becoming more of a problem to automobile dealers, because such a large percentage of car sales are made on a trade-in basis. The used cars must be reconditioned before they are resold, hence the range of repair work extends to many makes of car in addition to the particular line of cars that the dealer handles. The tool manufacturer has made every endeavor to provide adequate tools to take care of this situation. Now the automobile builder and his engineering department should give as much consideration as possible to the repairman's tool equipment. Standardized parts should be used and they should be assembled so that they can be taken apart and put together again quickly and cheaply, and so that one assembly can be removed from the car without disturbing other assemblies or parts.

GASOLINE FATTENS YOUR PURSE

THE steady workman has wheeled past the crowded tenement in which he used to live and has taken a little place out in the country, where his children can have fresh air and a garden. Usually he is buying his place. The slum that used to be considered inseparable from industry has all but disappeared and it no longer has anything to do with industry. People who live in slums today do so from habit. Industries are moving out of the cities and into the country, not only because of taxes, but because city industry cannot hire the sort of people it needs. Building a factory near a labor market no longer pays. A labor market is just a place where it is expected that a considerable number of men will be constantly unemployed and therefore the factory can shutdown and reopen at will. The modern factory cannot shutdown and reopen at will.

The change that all of this has made in retailing is revolutionary, and it extends back through manufacturing and the production of raw materials. We are in a new generation of business. The country store is being eliminated; the farmer drives past it into town, where he can get a better selection at lower prices. The town stores are dividing into three classes, department stores, chain stores and specialty shops. And they are all making money. The country store that is trying to do business in the old fashion is just shriveling up; the country store that is getting by has added a soft drink and ice cream parlor and carries only the little odds and ends for which it is not worth the farmer's while to go to town.

Consumption is at the highest point we have ever known; that is, we are far more prosperous than we have ever been. But those who do not understand the change that transportation has brought about are trying to buck the game, and they are getting hurt. It is advertising and not salesmanship that creates the demand in these days. The province of the salesman is narrowing and at the same time becoming more important. There is no longer any room for the man who drops in on a customer to inquire: "What can I do for you this month?" The merchant can buy cheaper than that agent can sell to him. The field for the salesman who can promote a new article or who can sell directly to the consumer is ever increasing. Mere order takers no longer pay.

The railroads are now giving certain service and in about half the time they formerly took. A number of the automobile companies, which formerly had to keep at least 30 days' requirements on hand to avoid being tied up, now work on a daily schedule and unload directly from the freight cars to the assembly floor, thus avoiding not only double handling but the maintaining of an expensive inventory. The Ford Motor Co. used to have a 90-day inventory; now its raw and finished inventory is all in process or in transit. It has a record of 10 days from raw material at the mines to a finished automobile sold in Chicago. The company is thus using about $300,000,000 less money than formerly.

The Pacific coast, where prices have always been high because goods had to be brought from the East across the Rocky Mountains, is now beginning to manufacture under the leadership of perhaps the ablest group in the Country. The coast has vast hydroelectric power, which is developing, and before long I expect to see its manufactured products exceed the value of its agricultural products. For it faces the largest potential market in the world—China.

There is no labor trouble of importance, and there cannot be, because men everywhere are getting good wages. What labor trouble we have, such as the recent strike in the anthracite coal fields, represents only the struggle of men far behind the times.

We are not going back to Normalcy in industry. We have come over the bridge from Normalcy, and the bridge has fallen down. What are the odds—will we keep going? From an article by Samuel Crowther in *Collier's Weekly*.

Seek Cracking Process for Production of Non-Detonating Fuel

By J. B. HILL[1] AND T. G. DELBRIDGE[2]

ANNUAL MEETING PAPER *Illustrated with* CHARTS

ABSTRACT

EXTENSIVE investigation is being undertaken by the petroleum industry with a view to producing, by improved cracking processes, a sufficient supply of internal-combustion-engine fuels that will fire without detonation at compression pressures of about 150 lb. It has been shown conclusively that gasolines produced by the present cracking processes are appreciably better from a detonation standpoint than uncracked gasolines from the same crudes and should have a compression limit of about 85 or 90 lb. It is also known from fundamental research work and from practical experience with cracking processes that higher cracking-temperatures tend to produce a larger proportion of the naphthenic and aromatic compounds that show non-detonating tendencies, hence it should be possible to work-out cracking conditions under which the reaction could be controlled so that the maximum production of the higher-compression fuels would be obtained.

Causes that led to the present demand for the greater engine-efficiency that results from higher compression-pressures are reviewed by the authors and they point out that changes made in gasoline-production methods to increase the supply of automotive-engine fuel have raised the boiling-point and the molecular weight of the fuel. This heavier fuel has a marked tendency to detonate under high compression and under extreme conditions of open throttle and low speed. The main problem, therefore, is that of producing a sufficient supply of fuel that will not detonate at the desired compressions. Every motor fuel, so far as known, has a certain compression-limit, under any given set of conditions, beyond which it will detonate, but the fuels differ in this characteristic and may be classed according to this limit.

Benzol and alcohol have very high compression-limits but the present supply of them is utterly inadequate, even when mixed with large proportions of gasoline to produce a fuel that will not detonate in engines at present in use. Benzol production is increasing rapidly and may become an important factor in the situation. Of a total production of approximately 10,700,000,000 gal. of gasoline in 1925, it is estimated that 2,000,000,000 gal., or 19 per cent, was uncracked gasoline from California and Gulf Coast crudes and had a compression limit of 85 lb.; that 5,200,000,000 gal., or 48 per cent, was uncracked gasoline from other crudes and had a compression limit of about 70 lb., and that the remaining 3,500,000,000 gal., or 33 per cent, was cracked gasoline having a compression limit of 85 lb. Thus, 52 per cent of the total, if segregated, would be non-detonating at 85-lb. compression. By mixing the production of 90,000,000 gal. of benzol with 90 per cent of the 70-lb.-compression gasoline, an additional 8 per cent of 85-lb.-compression fuel could be obtained. If one-half of the 500,000,000 tons of soft coal mined annually were subjected to the low-temperature carbonization process, the production of benzol would be increased to 1,250,000,000 gal. per year, which would be available for blending.

The production of gasoline by cracking is increasing more rapidly than total production and in 1927 may equal the production of uncracked gasoline. The petroleum industry looks to improvements in cracking methods as a solution of the problem of an ample supply of fuel that will not detonate in engines having a reasonably high compression-ratio, but it should know what compression the engine builders desire to use so that it can make a suitable fuel generally available and yet conserve the supply of fuel having a higher compression-limit.

IT has been recognized from theoretical and practical grounds, from the time of the inception of the Otto-cycle internal-combustion engine, that the higher the expansion or compression-ratio of the engine, up to reasonable limits, the higher would be the efficiency of the engine, other conditions being equal. At the time of the development of the automobile, however, gasoline was cheap and plentiful in this Country, and consequently there was little incentive to construct the most efficient engines at the cost of overcoming certain difficulties incident to the use of high compressions. Manufacturers, therefore, contented themselves with producing engines of simple construction which functioned uniformly and without any particular difficulties. The average compression-ratio 15 years ago was probably slightly under 4 to 1.

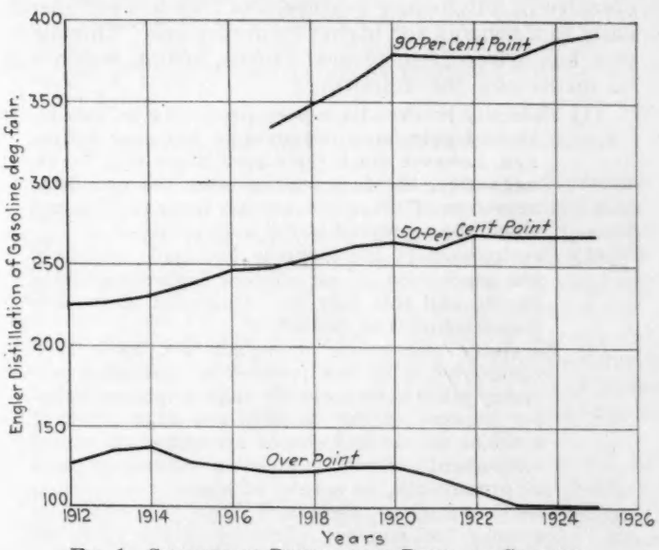


FIG. 1—CHANGES IN DISTILLATION RANGE OF GASOLINE
The Curves Show How the Distillation Range of Gasoline Has Risen Since 1912. It Should Be Noted That the Over-Point Has Decreased

Increasing popularity of the automobile, however, forced the petroleum industry to make a larger gasoline cut from petroleum to supply the increasing demand for gasoline, thereby producing a fuel that has a higher boiling-point than the gasoline for which automobile engines formerly were designed. Fig. 1 shows

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[2] Supervisor of process division, Atlantic Refining Co., Philadelphia.

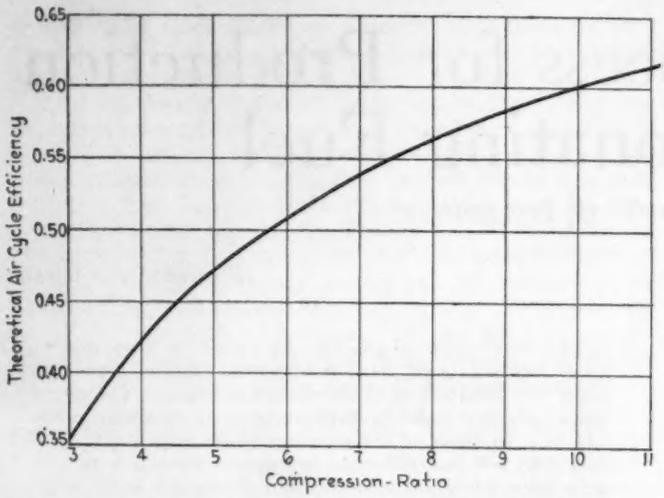

FIG. 2—EFFECT OF COMPRESSION-RATIO ON AIR-CYCLE EFFICIENCY
The Curve Represents Calculated Efficiency and Shows the Large Gain in Efficiency That Results from Raising the Compression-Ratio above the Present Average of About 4.45 to 1

the change in boiling range of gasoline due to these necessary changes in its method of manufacture. Passing over its various advantages and other disadvantages, this higher-boiling fuel produced the marked effect of knocking, or detonating, under the extreme conditions of open throttle and low speeds. A few automobile manufacturers who tried to increase their compression-ratios were forced by this property to return to their old and more inefficient ratios.

WHY HIGHER COMPRESSIONS ARE DEMANDED

Recently there has been a demand for greater efficiencies in automobile engines, and this has led specifically to a demand for higher compressions. This agitation has arisen from several causes, among which may be mentioned the following:

(1) Calamity howlers have been predicting an exhaustion of petroleum resources in the near future, and, however much their predictions may be exaggerated, the fact remains that the petroleum resources of the world are not inexhaustible and should be conserved so far as practicable

(2) Development of the airplane has made necessary the production of an efficient high-compression engine and this fact has stimulated automobile manufacturers to like effort

(3) European automobile manufacturers have been confronted with the problem of increased efficiency much more seriously than American builders because of the scarcity and high price of gasoline abroad and also on account of European methods of automobile taxation. European cars at present are, as a rule, of higher compression than American cars and show, therefore, a greater fuel economy. Inquiry by the public as to the reason for this state of affairs is beginning to make itself felt

For the foregoing reasons there has been an evident desire on the part of manufacturers in recent years to make use of higher compressions, and this has resulted in actual increases in compression-ratios in several recent models. A number of American cars are now being marketed with ratios of 5 to 1 or higher. This increase has been attained partly by the use of high engine-speeds and small cylinders.

The theoretical increase in thermal efficiency with increase in compression-ratio is shown by the curve in Fig. 2 in which the calculated indicated thermal efficiency, or air-cycle efficiency, is shown plotted against compression-ratio. The thermal efficiency is calculated from the relation

$$E = 1 - (1/R^{n-1})$$

where

n = the ratio of the specific heats, which is taken as 1.4
R = the compression-ratio

Sparrow[8] has shown experimentally that, while the actual efficiencies obtained in practice deviate from the theoretical as shown on the curve, the ratios representing the increases in efficiency by raising the compression-ratio are very close to the theoretical.

As against this obvious advantage of high compression, various disadvantages have been cited, such as rough running and hard starting. These objections are, to say the least, not insurmountable. They may lead, however, to the maximum economcial compression-ratio, and the opinion has been expressed by several authorities that a 7 to 1 ratio is the probable maximum.

CAN GRADE FUELS BY DETONATION TENDENCY

The real difficulty standing in the way of high compressions today is the tendency of gasoline to detonate under these difficulties. There is evidence that marked improvement can be made in avoiding detonation by following strictly mechanical lines, but the main problem is the production in sufficient quantity of a fuel that will not detonate at the compressions desired. Every motor fuel, so far as is known, has a certain maximum compression-limit, under any given set of conditions, and will detonate when that limit is passed. The term "non-detonating," as applied to fuel, is, therefore, a misnomer, as such a characteristic exists only with relation to the amount of compression. Fuels can, however, be arranged in the order of their detonating characteristics by determining the pressure at which detonation begins under a standard set of conditions. With the present rather similar designs of automobile engine, the so-called non-detonating fuels can be graded by the maximum compressions at which they will fire in the engine without detonation.

Because of this lack of a sharp distinction between detonating and non-detonating fuels, the fuel industry becomes closely interlocked with the automobile industry. It is to be expected that, until the production of high-compression fuels becomes as simple as that of low-compression fuels, the supply of the former will be conserved and that fuels will not be made any more non-detonating than the automobiles on the road demand. Conversely, it is to be expected that the automobile industry will not put cars of high compression on the road until a satisfactory fuel for them is easily and generally available. The fuel industry must know the degree of non-detonating qualities that the automobile manufacturers desire in a fuel before they can answer satisfactorily the question as to how much of such a fuel they can produce.

Various fuels that are not derived from petroleum have been resorted to in order to meet the demand for a sufficiently non-detonating gasoline for present automobile engines. Benzol is a thoroughly satisfactory fuel from this standpoint and apparently will withstand compressions of 400 lb. and more without detonating. Such compressions are not encountered, however, and, therefore, benzol can be diluted with gasoline to form a mixed fuel that will just meet present operating requirements. Even when used in this way, however, the supply of benzol is entirely inade-

[8] See National Advisory Committee for Aeronautics Report No. 205.

quate to meet any considerably increased demand for a fuel of this type. Similarly, alcohol ranks high on the non-detonating scale, but the production is insufficient, and alcohol has the additional disadvantages of solubility in water and of a low heat-value. The gasoline "dopes," such as aniline and particularly tetraethyl lead, are effective in raising the detonating compression of gasoline. Hence, as the demand for a fuel capable of withstanding high compression becomes universal, it seems to devolve upon the petroleum industry, because of the enormous consumption of motor fuel, to meet the demand with products from petroleum. The industry has attacked the problem resulting from this situation, and considerable research and production effort toward its solution is being made.

Detonation Tendency Increases with Boiling-Point

There are several distinct ways in which a fuel capable of withstanding higher compression-pressures might be produced from petroleum alone. It has been demonstrated repeatedly that the paraffin hydrocarbons that are the basis of normal uncracked gasoline show a tendency to detonate that increases as their boiling-point or molecular weight increases. For example, Ricardo's figures show that hexane will begin to detonate at a 5.10 to 1 compression-ratio and heptane at a ratio of 3.75 to 1. It would, of course, be possible now to make motor fuels of lower boiling-point for the sake of permitting higher compressions. Such a program, however, would so reduce the yield of gasoline from crude petroleum as to be wholly impractical. Thus, a normal Mid-Continent crude that is capable of producing 27 per cent of present motor gasoline without cracking, would, if run for aviation gasoline giving 50 per cent distillate at 221 deg. fahr. on the Engler test, produce only 10 per cent of this latter grade of fuel. The production of a high-compression fuel following this line consequently appears to be out of the question.

The second factor to be considered is that the uncracked-gasoline fractions from various crudes differ considerably in detonation properties. This variation is paralleled by a similar variation in the specific gravities of the gasoline fractions of any specific boiling-range. As a fairly general rule, the heavier the gravity the higher is the possible compression. Gasoline from Pennsylvania crude appears to be most likely to cause detonation, while the gasolines from most California and Gulf Coast crudes are among the least likely. Average gasoline from California crude appears to be equivalent in anti-detonating characteristics to a blend of about 10 per cent of benzol and 90 per cent of average Mid-Continent gasoline. It is certain that, by suitably segregating the gasoline from the crudes that are most non-detonating, a considerable production of a moderately high-compression fuel can be produced that will decrease detonation troubles in present cars very much and that will partially satisfy the popular demand for an anti-knock fuel. The compression-limit of such a fuel would be about 85 or 90 lb. The quantity of it that could be produced will be discussed later.

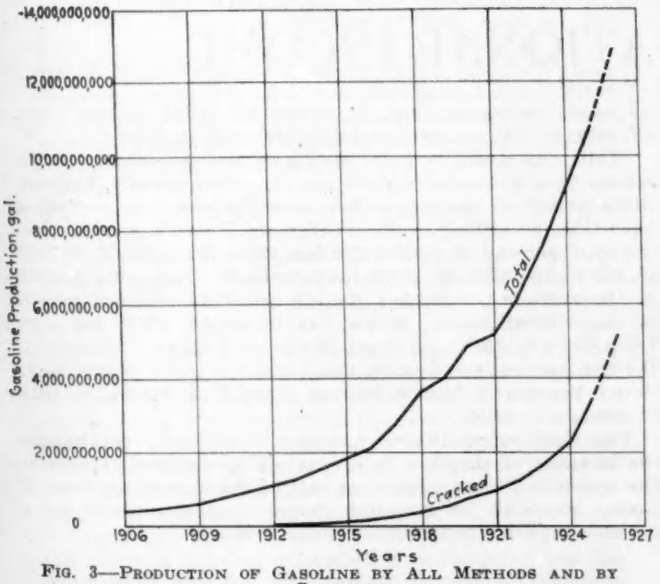

Fig. 3—Production of Gasoline by All Methods and by Cracking

The Rapid Rise in Production of Motor Fuel in the United States Is Shown by the "Total" Curve. The Lower Curve Shows the Rapid Increase in Production by Cracking Since 1915. The Figures from Which the Latter Curve Is Drawn Are Estimated

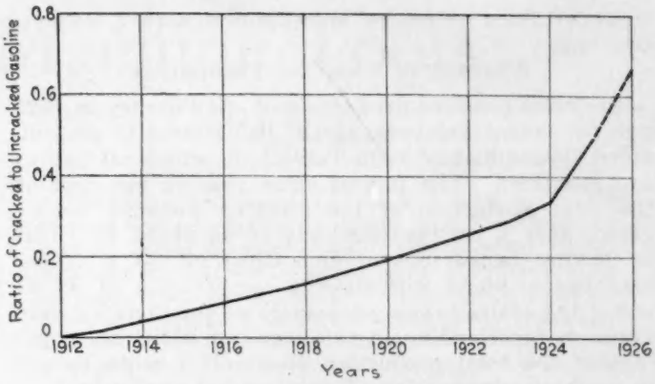

Fig. 4—Increase in Proportion of Cracked Gasoline

This Curve Is Calculated from Fig. 3 and Expresses the Change in Ratio of Cracked to Uncracked Gasoline and Shows More Clearly the Increasing Importance of the Cracked Product. The Ratio May Reach 1 to 1 in 1927

Cracked Gasoline Has High Compression-Limit

The third factor entering into a study of present high-compression fuels from petroleum is cracked gasoline. It has been shown conclusively that gasoline produced by present cracking processes is appreciably better from a detonation standpoint than uncracked gasoline from the same crude. As a result of this determination, cracked gasoline is no longer a material to be apologized for but has become a premium product. The average cracked product at present is about similar to the high-compression uncracked fuels and should have a compression-limit of about 85 to 90 lb.

This cracking experience is of great interest to the petroleum industry. It has been shown that the proportions of the various classes of compounds produced in cracking are dependent upon the conditions under which the cracking reaction proceeds. For example, it is known not only from practical experience with commercial cracking-processes, but also from fundamental research work that has been done, that higher cracking-temperatures tend to produce a greater proportion of naphthenic and aromatic compounds. Since both of these classes of compound show non-detonating tendencies and therefore are desirable, it should be possible to work-out cracking conditions for their maximum production and to control the cracking reaction by these conditions so that this maximum production would be obtained. Consequently the petroleum industry is looking at cracking as a possible means of producing fuels with a compression-limit of the order of 150 lb. and

is undertaking extensive investigation with a view to this end.

Analysis of Gasoline Production

The total gasoline production of the Country in 1925 was, in round numbers, about 10,700,000,000 gal., divided approximately as in Table 1, in which all figures are estimated. The figures show that 52 per cent of the total production of the Country would, if segregated, have a compression-limit of about 85 lb. This is leaving benzol out of consideration. If a 10-per cent benzol blend will allow a compression of 85 lb. and if the entire benzol production of the Country were available for blending in this way, an additional 8 per cent of the total production of gasoline could be put in the high-compression class.

TABLE 1—GASOLINE AND BENZOL PRODUCTION IN 1925, DIVIDED ACCORDING TO POSSIBLE COMPRESSION-LIMIT BEFORE DETONATION OCCURS

	Quantity, Gal.	Relation to Total, Per Cent	Compression Limit, Lb.[a]
Uncracked Gasoline from California and Gulf Coast Crudes	2,000,000,000	19	85
Uncracked Gasoline from All Other Crudes, Including Natural Gas Gasoline	5,200,000,000	48	70
Cracked Gasoline	3,500,000,000	33	85
Total	10,700,000,000	100	
Benzol	90,000,000		500

[a] Estimated.

While there is no immediate promise of any high-compression fuel from petroleum, the production of the moderately high-compression fuel is increasing. Fig. 3 shows the increases in total gasoline production and in cracked gasoline production during the last 15 years. It will be noted from the curves that gasoline produced by cracking is increasing at a more rapid rate than the total gasoline production. This fact is brought out even more strikingly in Fig. 4, in which the change in the ratio of cracked gasoline to uncracked gasoline is shown year by year. The curve is extrapolated to indicate the probable increase in cracking during 1926.

It has been indicated that the benzol production of the Country is practically negligible in comparison with the total gasoline production. A factor that may influence this condition in the future is the low-temperature carbonization of coal. It has been predicted that practically all bituminous coal that is mined will be subjected to low-temperature carbonization before shipping, and, as this process produces a yield of motor fuel of 5 gal. per ton of coal, the carbonization of even 50 per cent of the 500,000,000 tons of bituminous coal mined per year would furnish 1,250,000,000 gal. This fuel is high in benzol and would be available for blending with uncracked gasoline. It would, therefore, become an important factor to be considered.

Summary of the Present Situation

To recapitulate the foregoing and state the existing situation in the most condensed form

(1) The tendency toward higher compressions in automobile engines may be expected to result in increased demands for a fuel having a tendency not to detonate

(2) Resistance of fuels against detonation is a matter of degree, and the fuels may be graded according to the maximum compression-pressure they will withstand without detonation

(3) Fifty-two per cent of the total production of gasoline in the Country today is, if segregated, capable of use without detonating in engines of about 85 or 90-lb. compression

(4) The percentage of such gasoline available in 1926 should be appreciably greater than heretofore

(5) The petroleum industry is now exerting research energy toward devising cracking methods to produce a fuel having a much higher compression-limit.

TAXES AND THE NATIONAL INCOME

TWELVE and a half cents out of every dollar of the combined income of the American people for the year 1924 went into taxes, according to a study in taxation recently made by the National Industrial Conference Board, New York City. The total burden of Federal, State and local taxes has nearly doubled in proportion to national income since 1913, rising from 6.9 per cent in 1913 to 12.5 per cent in 1924; also showing an increase over the preceding year, 1923, when the total tax burden amounted to 11.8 per cent of the national income.

While the increase of taxation in proportion to national income in 1924 in part was due to increased State and local tax levies, it was in part, however, also due to a decrease in national income for that year. The national income in 1924 is estimated by the Conference Board at $63,000,000,000, as against $65,000,000,000 in 1923. Expressed in dollars of current purchasing power, the national income in 1924 was almost double that of 1913; deflated to take into account the decline in purchasing power of the dollar since 1913, the national income in 1924, in terms of "1913 dollars" was $39,000,000,000, as against $32,000,000,000 in 1913.

While the average gain in population, according to the census figures, has been about 1½ per cent annually, Federal, State and local taxes together have increased at so fast a pace that, according to the Conference Board's computation, the total per capita tax burden has risen from $22.73 in 1913 to $70.97 in 1924, or more than trebled. The same is true of the average tax burden of each gainfully occupied person in the United States, which has increased 208.8 per cent from 1913 to 1924, or, from $59.25 to $182.94. Measuring the tax burden per family, the social unit, the Board finds that it has nearly trebled, having arisen from $102.12 in 1913 to $304.23 in 1924.

The Conference Board, however, specifically emphasizes the increase of taxation in proportion to national income as the truest and most significant test of the burdensomeness of taxes, inasmuch as national income is the measure of a nation's capacity to spend.—*Economic World.*

How to Select Machine-Tools

By W. G. Careins PRODUCTION MEETING PAPER

ABSTRACT

THE selection of machine-tools is largely a matter of judgment, based on the consideration of many variable factors. No fixed rules can be laid down, but the uses to which the machines are put are divided roughly into three classes which govern to a large extent the type of machine that should be purchased, whether they should be machines of a wide range of usefulness, standardized machines equipped with special tooling or special-purpose machines of special design capable of very large and continuous production. To determine into which of these classes the requirement for new machines falls, an analysis should be made of the following factors: (*a*) quantity of production required and its duration, (*b*) method of machining and tolerances and finish required, (*c*) possibility of a change in design of the product, (*d*) cost of production, (*e*) when delivery of machine is required, and (*f*) money available for the purchase.

When the type of equipment needed has been decided, an inquiry to elicit offers of machine-tools and quotations of prices may be broadcast through trade journals or addressed to a selected list of machine-tool builders. The latter method is preferable, perhaps, as it usually brings representatives of firms that build other types of equipment as well as the particular type desired. Information given in the inquiry should be as complete as possible and in the case of a special machine should be accompanied by a rough sketch and a statement of the method of handling the work into and out of the machine, surfaces to be machined, work that has been performed in previous operations, hardness of the material and data on feeds and speeds, and, if possible, a sample of the work to be performed. The builder should be given as much latitude as possible in making his recommendations, because the buyer is dependent largely upon the manufacturer for information regarding new methods and processes that have been developed among machine-tool users.

Some machine-tool builders, when making quotations, give only a general description of their machine and the price and, if any production estimate is given, merely state the number of pieces per hour that may be expected without accompanying data to support the statement. Other builders send very complete information and guarantee a definite production under certain standard conditions at a definite cost. It is a simple matter for the buyer to check the information given in the latter kind of quotation and compare it with other quotations.

Production capacity, cost of the equipment and its ability to give satisfactory service are major factors to be considered when making a selection of machine-tools. Other factors that may be of much or minor importance according to circumstances include proximity and reputation of the builder, service that may be expected, completeness of the builder's stock of repair parts, whether or not the builder furnishes repair-parts lists, floor space occupied by the machine, and convenience of its operation. Another factor of considerable importance is the question of standardization of equipment in a production plant. If a number of machines of one make, type and size are in use in the plant, it is often advisable to continue to buy the same make and size of machine, so that attachments and fixtures will be interchangeable and the stock of repair parts will not need to be increased.

The author concludes his paper with brief references to some common causes of trouble with machine-tools, such as inadequate countershafts, electric-motor mountings that are apparently added as an afterthought, insufficient width of belt pulleys, inadequate oil-pumps; cast-iron heads, feed mechanisms, cams and levers; lack of accessibility for making repairs, and multiplicity of oil-cups and oil-holes instead of a system for lubricating all parts from a central reservoir.

THE selection of machine-tools is not an exact science governed by fixed rules. It is largely a matter of judgment determined by proper consideration of a large number of variable factors. Since judgment is a mental process, it is impossible to eliminate the human element in specifying machine-tool equipment.

Certain factors assume more importance in the mind of one person than they do in the minds of others. One person may be influenced largely by price, another by past experience, and still another by some intangible thought that it is impossible to explain satisfactorily. It is highly possible that if five persons were asked to purchase a 16-in. lathe for the same job, each would specify a different make than the others and, in his own mind at least, would have a perfectly logical reason for his selection.

It is not the intention, however, to deal in this paper with the complexes of machine-tool buyers, but to attempt to analyze some of the more concrete factors that should be given consideration in the selection of machine equipment.

THREE CLASSES OF MACHINE-TOOL USE

The uses to which machine-tools are put in various plants can be divided roughly into three classes, each of which places somewhat different demands upon the equipment. The first of these uses is found in plants that are devoted to the manufacture of a variety of products in which duplication occurs to a very slight extent, in the maintenance and tool-departments of production shops and in the small job-shop. This class of manufacturing requires tools of the widest possible range of usefulness, size and type. The factors governing the selection of equipment for this type of service are extremely variable, as every purchase requires special analysis.

The second class of use is found in shops devoted to fairly large production of a product that is subject to frequent changes of design, fluctuations in seasonable demand or other conditions that make it impracticable to assume that manufacturing conditions will be settled for any long period. The average automobile-shop of 5 or 10 years ago was a good example of this type of manufacturing. The demands on machine-tools in this class of service are not nearly so varied as in the previous class, although the equipment must still have a certain degree of flexibility. The machine equipment in such a plant would consist very largely of standard machines equipped with special tooling, the standard machines being supplemented with an occasional single-purpose machine to take care of certain special or key operations.

[1] M.S.A.E.—Production engineer, Ajax Motors Co., Racine, Wis.

The third class is found in plants engaged in large and intensive production of a standardized product that is subject to very little change in design from year to year, and in which standards of quality and the production methods to produce these standards have been thoroughly established. In plants devoted to this type of manufacturing will be found many machines of special design, often automatic or semi-automatic, which are capable of giving extremely large and continuous production. Standard machine-tools from which have been removed all accessories and attachments that are not needed to perform satisfactorily the work for which they were purchased will also be found. The machines in this class of service have been selected and arranged so that one operator can take care of two or more machines with the minimum of effort. In fact, in some cases the machines in a production line have almost lost their identity as individual machines and have become, through the use of conveyors and automatic handling-devices, an integral part of a production unit that takes raw material at one end and turns out a finished article at the other.

Factors To Consider in Selection

When new equipment is needed, it should be the first consideration of the person or persons responsible for its selection to analyze the factors in the situation along the following lines and determine into which of the three classes the requirements fall:

(1) *Production.*—Ascertain the production required and, as nearly as possible, estimate the period during which this production will be maintained; also consider the possibility of a gradual or rapid increase in demand. Remember that, if a special machine is purchased which has capacity to take care of the entire requirements and this machine breaks down, the production stops

(2) *Methods.*—Give very careful consideration to the method of machining to be used, the inspection tolerances and the quality of finish required. If the equipment is for expansion or replacement and it is deemed advisable to change the methods, care should be taken to determine the effect of this change upon previous or following operations

(3) *Design.*—Investigate the possibility of a change in design that would make special machines or expensive tooling obsolete. Some automotive parts, such as manifolds and brake rigging, are particularly subject to frequent changes of design

(4) *Production Cost.*—Special machines usually are selected with the idea of reducing costs. Production costs are calculated from direct labor, depreciation, interest on investment, maintenance, floor space, and other items of this nature. However, in estimating the cost of producing a part on a special machine, the depreciation factor should be approximately twice as great as the depreciation factor that is used for standard machines, because the special machines are likely to become obsolete, as a result of changes of design, before they are worn out. A higher maintenance cost should also be figured because of the difficulty and expense of obtaining repairs, and also because defects in design may make excessive repairs necessary

(5) *Delivery.*—If equipment is needed at once, the possibility of purchasing special equipment is eliminated, as it is very seldom that a special machine can be designed, built, delivered, and put into successful operation in the plant in less than 3 months, and often a much longer time is required. A method of overcoming this difficulty is to purchase standard machines that can be converted readily to other work at a later date when the special machines are purchased

(6) *Budget.*—The amount of money available for the purchase of new equipment often has considerable bearing on the selection, forcing the purchase of standard equipment even when more expensive special equipment is desirable

How To Determine What Machine To Buy

Assuming that a decision has been reached regarding the type of equipment that will be purchased, the next problem that confronts the buyer of machine-tools is to determine what particular machine to buy. In the first place, the inquiry must be conveyed in some way to the machine-tool builders. This can be done either by "broadcasting" the requirements through trade papers or by direct inquiry addressed to a limited number of builders whose machines it is desired to consider in making a choice. The latter method is preferable, perhaps, since a general inquiry for one type of machine usually brings a large number of representatives and salesmen from firms manufacturing other types of equipment, the assumption evidently being that if lathes are needed milling machines may also be needed, or, if gear-cutting equipment is required, machines will also be required for turning gear-blanks, all of which causes needless expense both to the seller and buyer. In the case of special machinery, if some question arises as to possible sources of supply, it is good policy to investigate similar machines in other plants, noting the good points of each and the satisfaction they have given.

The information given to the machine-tool builder when making the inquiry should be as complete as possible, so that he can make his recommendations intelligently. If special machines are wanted, he should be provided with a rough sketch of your ideas of the machine, indicating the points of location, and with information as to the production per hour required, method of handling the work in and out of the machine, surfaces to be machined, work that has been performed in previous operations, hardness of the material, and any data that may be available regarding feeds and speeds that have been used in the past on this operation; also any other general information available and, if possible, with a sample of the work to be performed. Much the same procedure should be followed in the case of standard machines except that it probably would not be necessary to make sketches.

In all cases the machine-tool builder should be given as much latitude as possible in making his recommendations, as it is to the advantage of all concerned that the wide experience of the builder be utilized to the fullest extent. In fact, the buyer is largely dependent upon the tool manufacturer for information regarding new methods and processes that have been developed among users of machine-tools.

Complete versus Meager Quotations

The work of the buyer is now temporarily completed. The next step must be taken by the builder, whose task is to analyze thoroughly the data furnished, request further information if any point is questionable and, from this analysis, to make his recommendations. A few companies, upon the receipt of an inquiry containing the information mentioned, simply make out a formal quotation giving a general description of their machine,

attach it to a circular and forward it to the buyer. If they give a production estimate at all, they probably state that a certain number of pieces per hour can be expected, but give no data as to how this estimate was arrived at, no data on feeds or speeds used and no information as to whether the figure given was 100-per cent time or average floor-to-floor time; in fact, practically all the information given is that they have a machine to offer at a certain price and promise delivery at a certain date. A proposal of this nature is very unsatisfactory, as it always leaves a doubt in the buyer's mind as to whether the builder has neglected to furnish this information through carelessness or has omitted it intentionally, knowing that complete data would not show up his equipment to the best advantage.

On the other hand, proposals from other builders will be received in neat and attractive form, giving a complete analysis of how the work would be performed on their equipment, guaranteeing a definite production under certain standard conditions, specifying the quality of work to be expected, and including a sketch showing the tooling recommended, and, perhaps to substantiate their claims, submitting time-studies taken on identical or similar work performed on their type of equipment in other shops. In short, instead of merely offering a machine-tool at a certain price, they offer guaranteed production at a definite cost. It is a simple matter for the buyer to check the information given in such a quotation and compare it with other quotations received.

Major Factors To Be Considered

The three items that probably would be of most interest to the buyer when comparing quotations are production, cost of equipment and ability of the machine to give satisfactory service. Knowing these three things, it is possible to approximate closely the cost of producing work on any machine, as the direct labor-cost can be calculated from the prevailing labor-rates and the number of pieces produced in any given time, while the fixed overhead expenses directly chargeable to the machine are determined largely from the original cost of the machine, the repairs and its estimated life. The machine's production and its cost are fairly definite, but the third factor, life and maintenance cost, must be determined largely from experience and the policy of the buyer's company in regard to replacing equipment. Some firms prefer a reasonable length of machine-tool life with the minimum of repairs, others prefer to have a much higher maintenance-cost and to get the maximum of service from their equipment before it is discarded. The life of a machine is dependent upon its type as well as upon maintenance. A complicated machine having a large number of wearing parts, such as gear-cutter equipment, cannot be expected to give the same length of satisfactory service as a simple machine, such as an engine lathe, having no complicated mechanical movements.

Many other factors should, of course, be considered before making a final decision. One of the most important, perhaps, is the reputation of the builder, not only as regards the design, workmanship and reliability of his equipment, but also the service and help that the user may expect from the builder after the machine is in operation. Does the builder maintain a complete stock of repair parts or will the machine have to remain idle, in case of a breakdown, until the necessary parts can be made either in the buyer's or the builder's plant? Does the builder furnish repair-parts lists so that repairs can be ordered intelligently and, finally, does he try to help the user to obtain the maximum usefulness from his equipment through the medium of service bulletins and expert demonstration?

Other Considerations Sometimes Important

Floor space occupied by the equipment under consideration is sometimes an important item. A large drum-type continuous-milling-machine occupies approximately 25 per cent of the floor area that is covered by a planer-type milling-machine of equal production capacity. If a plant is situated in a locality where real-estate values are very high, the item of floor space might assume sufficient importance to be a major factor in considering production cost.

Often the remark is made that a certain machine is "very satisfactory in every way except that it is so clumsy to operate that our workmen much prefer other makes of machine." This statement is often true, in fact, machines are on the market today that require so much waste effort on the part of the operator that their production capacity is reduced considerably. A good illustration of this is the average radial-drilling machine. When an operator has several holes of various sizes to drill, it is necessary for him either to drill all holes at one or two speeds or to walk around to the back of the machine and change the speed through the gearbox. On certain operations this question of control assumes considerable importance.

Although appearance is not a factor that should have an important bearing on the selection of equipment, it is impossible for the average person not to be influenced to some extent by symmetrical design and neat and attractive finish.

Location of the machine-tool building plant also sometimes has some bearing on the final selection. If two machines are offered, one of which is built close to the buyer's plant and the other at a distance of several hundred miles, and if other factors are equal, certainly the machine built near-by should be chosen, because of the ease with which repairs and service help can be secured.

Another factor of considerable importance in the final selection is standardization of equipment. If several machines of a particular make are already in use in a shop, it is often desirable from the standpoint of economy to continue to buy that make and size of machine, so that all attachments and fixtures can be interchanged from one machine to another and the stock of repair parts carried in the plant need not be increased. In the case of milling machines, for instance, the dividing heads, arbors, rotary tables, and other parts can be shifted from one machine to another, and, in the case of turret lathes and automatic screw-machines, the same stock of cutter holders, roller turners, chucks, and many other standard and special tools can be made to serve a large battery of machines. On the other hand, if the equipment used is of a number of different makes, it may be necessary to carry a complete assortment of tooling for each make of machine. Hence, standardization becomes of great importance in some types of manufacturing.

Some Common Causes of Trouble

Assuming that, after proper consideration of all the foregoing factors, a particular machine has been purchased and is in operation in the plant, a few of the common weak-points in design and workmanship that are likely to give trouble may be mentioned. First, many machines are first-class in every respect except that the manufacturer insists upon using a countershaft that in no way measures up to the standard of the

machine or, if the machine is arranged for direct motor-drive, the motor mounting is apparently an afterthought in the design. In some cases the width of drive pulleys for belt drive or belted motor-drive evidently has been calculated for belts running under ideal conditions, which conditions are seldom met in the average shop.

Another common cause of trouble on machine-tools is the pump for the circulation of cutting lubricant. Nothing is so exasperating in the machine-shop as pumps that have to be primed every time the machine has stood for some time or that do not have sufficient capacity at all speeds.

Cast-iron gears in the head or feed-mechanism are almost always a source of trouble in machine-tools, and the same is true of cast-iron cams and levers.

Many manufacturers still design their machines with a large number of oil-cups and oil-holes, instead of lubricating all parts of the machine from a central point so far as possible. As many repairs are made necessary because of lack of lubrication as from wear and accident. It is possible to induce the average mechanic to keep a central reservoir full of oil, but not many mechanics will properly lubricate a machine if they have to oil 15 or 20 different points once or twice a day.

Lack of accessibility for making repairs is another serious fault. Often it is necessary to dismantle a machine almost completely to make some minor repair or adjustment, and this is made still more difficult because few manufacturers furnish instructions for taking-down the machines or blueprints showing the construction of their machines.

THE DISCUSSION

CHAIRMAN EUGENE BOUTON[3]:—The common practice among some of the machine-tool builders in regard to the quotation estimate is to ask questions pertaining to the job that do not enter into the estimate at all, such as "How many pieces do you want to produce per day?" "What speed and feed are you running now?" or "What type of equipment are you using?" When they are given a drawing and perhaps a sample of the work, they should develop their estimates from those. Half of the inquiries on the questionnaires that are sent by some of the machine-tool builders are not answered, or they should not be answered.

While the machine-tool builders have made wonderful improvements in the lubrication features of machines, much still remains to be done in that direction. In setting production standards, the common practice is to make allowances for the operator to oil the machine at frequent intervals. However, machine-tool operators are very negligent in this respect. Machine-tools that have mechanical means of lubrication will be lubricated more thoroughly than by relying on the operator to do so with an oil-can or other hand method. Automobile builders have also recognized the fact that oil-cans and grease-cups are not very reliable lubricating mediums and some manufacturers have installed centralized lubricating devices that by pressing a pedal or moving a lever will thoroughly lubricate the entire chassis.

JOSEPH LANNEN[3]:—In our plant recently we have had to face this condition. We started to see what could be done in the way of cleaning up our machine-shop. First, we hired tinsmiths to make guards for the machines. Some of the parts for which guards were made would be impossible for the machine-tool builders to cover, but in a great number of cases no means were provided whereby we could keep the chips and shavings from falling on the floor, so we made guards or pans to catch them. We also made arrangements to keep oil from dripping on the floor. Sawdust alone cost us $180 last month. We tried to cut-down that cost.

I do not see why, with a little thought on the part of the machine-tool builders, a certain amount of the shop-dirt could not be eliminated. Perhaps the machine-shop is supposed to be a dirty place. It has been said that if you buy a horse you must expect to clean the stable, but we cannot buy the horse according to specifications.

BUILDERS SHOULD STANDARDIZE MACHINES

JOHN YOUNGER[4]:—The question of standardization is of interest to me. In the early motor-truck days, when ¼, ⅝ and 1¼-ton trucks and all through the range of sizes were built, if a man wanted a truck of a certain size he could buy it. Today, if he wants a truck of a different size than is built, does he get it? He does not; he gets a standard-size truck, a 1, 2, 3, 4, or 5-ton truck. He must take those sizes because the builders of trucks conferred and said, "We will build only those sizes and no others."

As I see the situation, the machine-tool builders should get together and say, "We will standardize, and if a man want a 21½-in. swing-lathe he will not get it." That is the way the Society has undertaken standardization. For instance, spark-plug threads are a certain size, because the spark-plug manufacturers cooperated and standardized the threads. I think it will help the machine-tool builders if they will cooperate and establish standards.

STANDARDIZATION UP TO BUYERS

E. F. DUBRUL[5]:—That is true and I accept the suggestion. The truck builder and the spark-plug manufacturer were more progressive business men than the machine-tool builder. The latter is not strong enough to hold out for the price that he should receive for his product. He sells large-production, well-patented machines on a cost plus basis, instead of on a basis of utility to the user.

The machine-tool customer knows something about producing the things that are made in machine-shops. He knows the cost of labor and material but conveniently forgets many other things, such as experiments, obsolescence and the many other concealed costs. When he talks to the machine-tool salesman he tells him that the machine costs too much. The salesman is a poor businessman and, not being trained to laugh at this nonsense, goes back to the office and tells the executive about the "hard-boiled customer," and the executive, who is not a very good merchant, reduces the price. Some very unfair arguments are used by the skillful buyers who are employed for the purpose of saving money and who try to buy everything at rock-bottom prices. When they deal with a man who is more skilled in mechanics than in these business matters they advance very childish arguments.

A man who wants a special machine can easily find some machine-tool builder who is a poor enough merchant to produce it for him at a price that is far below its true value. Not every machine-tool builder has learned to sell "production, not products."

After 5 years of effort to establish standardization from inside the machine-tool industry, I have concluded

[3] M.S.A.E.—Supervisor of time study, Chandler Motor Car Co., Cleveland.

[4] Tool supervisor, Paige-Detroit Motor Car Co., Detroit.

[4] M.S.A.E.—Publisher, *Automotive Abstracts*, Cleveland.

[5] General manager, National Machine Tool Builders Association, Cincinnati.

that it must come from the outside. Users will be able to get standardized machines just as soon as they really demand them. As it is now, the machine-tool builder is using up too much of his capital in putting out unstandardized machines. If the buyer does not pay enough to make the business attractive, not enough producers will be available to supply the wants of the automotive industry in the future.

CHAIRMAN BOUTON:—I want to support Mr. DuBrul's explanation by an instance that I think it is not out of place to mention here. K. L. Herrmann, of Detroit, a very active member of the Society, wanted a hobbing machine built about 3 years ago but could not get it. The Studebaker Corporation of America, with which he is connected, built one to prove its theory regarding the hobbing of gears, and some time thereafter a machine-tool builder was induced to construct 20 of those machines. That incident has resulted in some wonderful improvements in hobbing equipment.

ELIAS N. SAWYER[6]:—I should like to ask if Mr. DuBrul does not think that the S.A.E. Standards are not so much the result of the thought of the executives but come from the men in the machine-shop or plant. I believe that the men in the machine-tool industry are made of the same sort of clay as the men in the automotive industry. They are not supermen but I believe that the majority of the standards originated with the engineers and automotive men and not with the executives and buyers and salesmen.

BETTER MACHINE SALESMEN NEEDED

MR. DUBRUL:—I believe that is true. I think our engineers should be more active in standardization. But when I call a meeting of engineers to make recommendations, our executives find that the cashbox does not permit of their attending the meetings because they have not charged enough for the machine-tools in the past.

The ordinary buyer of an automobile does not know much about what he is getting but the buyer of machine-tools does know and the machine-tool salesman is selling to the engineering brains of the world. A type of man is required like the one Mr. Careins has described, who can show the different points of his machines and make his proposition to the buyer. Some of that type are in business today, but not as many as we need. I am in favor of paying them the money that will make it worthwhile for that kind of man to come into the machine-tool industry because that type of salesman will perform a first-class service. The user will, of course, pay more, as he should, for better selling service.

I have had a hope that someday the machine-tool industry would have a staff of experts who know all about milling, a staff of drilling experts and one of turning experts, men who would be consulting production men, engaged by the National Machine-Tool Builders Association. This is not an impossible dream. Some 400 engineers in all parts of the Country are engaged in showing customers how to use cement, so why not a similar staff to educate users how to use machine-tools? I hope to see the time when we shall have men who can inspect a shop and tell what type of machine a plant should be using, and all that. Such men could show our industry's customers how to save considerable time and money. In the long-run we could furnish much better machine-tools because those men would be studying production. We have many such dreams but do not get money enough in the prices to carry out these dreams.

TRADE SOON SWINGS OVER TO STANDARDS

R. R. POTTER[7]:—One reason the machine-tool builders are not more enthusiastic about standardization, I believe, is that they have clearly in mind the difficulty of going to customers who have many machines with the present design of spindle nose or lead screw or whatever it may be and selling them something different. If the machine-tool builders could cooperate in some way through the Association and, if necessary, borrow the money so that they could afford to establish a set of standards, they would be surprised how quickly their business would become standardized. I think that in 10 years the business would swing around to that set of standards. I would like to illustrate what I mean by citing a personal experience. About 10 years ago, when I was connected with a company that was making universal-joints, the S.A.E. Standard on taper connections or fittings was approved. At that time the industry was using all kinds of fittings, different diameters, different lengths, everything was different. A manufacturer would start with some special design and that had to be followed. When the S.A.E. Standard was announced, this practice did not cease at once but whenever a new company started to lay-out a model, or when a new design was made by an established company, they used the S.A.E. Standard. It was surprising how quickly, perhaps in 3 or 4 years, the great bulk of our business was on S.A.E. Standard tapers. I believe the same result would work-out with machine-tools, more slowly, perhaps, but reasonably soon.

MONEY NEEDED FOR STANDARDIZATION

A MEMBER:—It is surprising to learn how much money is given in the East to the large universities as endowments. Perhaps it would be possible to induce some rich man to endow a standardization foundation; a $1,000,000 or so for this particular work would do wonders. The money is needed badly. All that is required to bring about a standardization program is money.

FORREST E. CARDULLO[8]:—Some one here has suggested that the machine-tool builders should borrow, or persuade some rich man to give, $1,000,000 or so to develop new machine-tools and processes for the benefit of the automotive industry. Now, think for a minute. Who pays the cost of experimental and development work in the automotive industry? The customer, and only the customer, pays for it. Why should not the automotive industry, as the principal customer for machine-tools, pay for the experimental and development work that must be done by the machine-tool industry, to bring out new tools and processes for the automotive industry's needs?

You are looking at this question from an entirely different standpoint, financially, than the machine-tool men. You are connected with an industry that is growing continually. The number of cars that are being produced in this Country is increasing steadily year by year and you still have a great many potential customers. You have not yet reached the so-called "saturation point." On the other hand, the machine-tool builder is always producing for a saturated market.

AUTOMOTIVE INDUSTRY MUST PAY THE PRICE

Imagine what would happen to the automotive industry if it were in the position of the machine-tool industry, and every automobile sold were completely worn out in service by a man who was a careful driver and who

[6] M.S.A.E.—Works manager, Rollin Motor Co., Cleveland.

[7] M.S.A.E.—Assistant to president, Shakespeare Co., Kalamazoo, Mich.

[8] Chief engineer, G. A. Gray Co., Cincinnati.

would always keep his car in good condition. Also assume that every customer is hard-boiled and demands to be shown before he buys, that the automobile will earn 100 per cent a year in his business. When such a time comes, the financial condition of the automotive industry will be somewhat different from what it is now. The profits will not be so large and the attitude toward the costs involved in the development of new ideas will be different.

The customer has to bear the cost of development in every industry, or the industry will have to go out of business. No reason exists why the automotive people should expect the machine-tool builders to play Santa Claus for them. The general attitude of the automotive industry when purchasing machine-tools is that they want the best tools for the least money. Unfortunately, one cannot buy the best tools for the least money. Occasionally, a company can be induced to design and develop a good tool, and then some one who has not borne the expense of development can be persuaded to build a passable substitute on a narrow margin of profit. Such a policy is neither good ethics nor good business. It discourages the progressive firm, and so kills the goose that lays the golden eggs. You will have to make up your minds to pay for what you get, and, until you are willing to pay for what you want, you will be unable to get it.

Another unfortunate idea that has been revealed by the discussion this afternoon is the notion that the machine-tool man should do the production engineering for his customers, free of charge. Production engineering is the job of the customer. The machine-tool builder tries to make the best type of machine-tool he can build in reasonable quantities for a given price, but he cannot be expected to develop a special machine-tool for a specific operation and supply two or three of them at a comparable price. The market is too narrow.

How Far Automotive Standardization Goes

Another question that has been discussed is just how far standardization actually has progressed in the machine-tool industry. In the automotive industry, it has progressed only so far as it has saved money for the industry itself. Otherwise it has not progressed at all. I have driven a car for a number of years but, if I buy a new car of a different make, I am likely to find that the gearshift is different from the one I have been using. A standard gearshift is on the market but not all of the companies in the automotive industry have adopted it. If I want to buy repair parts for my car, say a new universal joint to replace a worn one, or a new timing-gear, I must buy it from the company that builds that particular car.

Tires and spark-plugs have been made interchangeable, but that is because their standardization saves money for the industry. You have carried standardization only to the point that suits your pocketbook, but you are asking the machine-tool builder to carry it much farther than that. If you can make it worthwhile, and save the machine-tool industry money by a standardization program, you will find us following that program just as far as the automotive industry carries its standardization work.

Chairman Bouton:—The initial cost of machine-tools has something to do, of course, with the selection of machine-tool equipment, but it is the ultimate saving in direct-labor that determines whether or not the machine-tool at that price would be a good investment.

E. P. Blanchard[9]:—I am interested in the analysis of Mr. Careins's paper for he so clearly indicates that machine-tools must be selected for maximum economy under the conditions that exist in each shop and the various demands for quantity and quality production. Conditions exist in our own shop where a machine-tool is busy only 3 months in the year, yet the labor saving and the accuracy of its work when compared with older methods that require full-time operation have been sufficient to make that tool worthwhile. Only part-time use of equipment may still prove to be ultimate economy.

Some remarks made by A. J. Baker at the Production Meeting in Detroit 3 years ago brought out very clearly that the selection of tools is a matter of production cost. He pointed out then[10] something that many of us have learned since, that floor space is not a big factor except where the shop is in a high rental district and additional building is entailed. Then, of course, it is of much importance.

As for the scoring of machine tools, the manufacturers of automotive products are looking for certain features essential in production. Let them state those features and standardize them; then, in selecting, score a point for tool application, score a point for control of the machine and so on. Machine-tools can be standardized to your requirements on that basis.

Confidence and Cooperation Needed

I have had considerable experience with special-purpose machines and here is a typical quandary: we are asked to build a special machine for a special purpose—just one machine. The machine is built for the purpose and it partakes of certain standard units in our regular models. When it was offered to the customer, the first criticism made is, "you have too much in it. We do not need all of your feeds and speeds nor do we require this feature and that." But it is cheaper for us to build the machine that way than to design and build an entirely different model; yet if we stripped down to just the essential features for the job, the cost of special work would bring the price considerably higher. We are caught either way; if we offer the machine as it is, it is too much machine for the job and if we offer less machine at a higher price, the price is wrong.

A little more confidence and cooperation, talking over facts and costs together and with those in other industries, would benefit all of us. Salesmanship training is needed in the selling of machine-tools, but it is so much more an engineering problem that the closer we can get together the better.

[9] Advertising and assistant sales manager, Bullard Machine Tool Co., Bridgeport, Conn.

[10] See The Journal, December, 1922, p. 526.

PROPOSED AMENDMENTS TO SECTION CONSTITUTION, BY-LAWS AND RULES

THE necessity for revising the standard Section Constitution, By-Laws and Rules was one of the topics considered by the Sections Committee at its meeting in Detroit on Jan. 26, because the Section Constitution as it now stands is practically obsolete. A redraft of the Constitution was submitted for discussion and, in accordance with the decision of the Committee, the proposed revision has been sent to each Section for action. The suggested amendments will be proposed at an early meeting of each Section and, after being duly seconded, will be submitted for discussion at a subsequent business meeting of the Section.

The portions of the Section Constitution and By-Laws which it is proposed shall be amended are given below. The matter in brackets represents that which will be omitted if the proposed amendments are adopted. Suggested additions or substitutions are printed in italics.

SC 6 [There shall be only one class of membership in the Section, to which any individual member of the Society who is in good standing shall be eligible. Affiliate members, affiliate member representatives and enrolled students shall not be eligible for Section membership.]

Any individual member of the Society who is in good standing shall be eligible to membership in the Section. Affiliate member representatives and enrolled students shall be eligible for Section membership upon formal application and shall be entitled to all Section privileges except voting and office-holding power. Such application shall be subject to the approval of the Governing Committee of the Section.

SC 7 Each member of the Section, *except affiliate member representatives and enrolled students, who is* present in person at any meeting shall be entitled to one vote on any question submitted except that no one who does not hold voting power in the Society under the Constitution thereof shall vote in the annual election by the Section of a member of the Nominating Committee of the Society, as provided in the Constitution of the latter.

SC 9 [Applications for membership in the Section shall be made in accordance with the requirements of the By-Laws of the Section, and shall be subject to the approval of the Governing Committee of the Section. Such approval shall constitute election to the Section upon payment of dues.]

Individual members of the Society shall become members of a Section upon the payment of Section dues to that Section. Approval by the Governing Committee of the application for membership in a Section of an affiliate member representative or an enrolled student shall constitute election to the Section upon payment of dues.

SC 11 A member whose dues to the Section remain unpaid for 3 months shall be considered as not in good standing in the Section and shall, *in the discretion of the Governing Committee,* not be entitled to vote [therein] *or enjoy any other privileges accorded Section members in good standing* until dues in arrears have been paid in full.

A member whose dues remain unpaid for 1 year shall cease to have any further rights in the Section and his name [may] *shall* be stricken from the roll of the Section *unless different action is recommended* by [action of] the Governing Committee.

SC 17 [The administration year of the Section shall be fixed by the Section and specified by its By-Laws. The fiscal year of the Section shall be its administration year, unless otherwise provided in the By-Laws.]

The administration year and the fiscal year of the Section shall be as specified in the By-Laws.

SB 5 The officers of the Section shall be a Chairman, a Vice-Chairman, a Secretary and a Treasurer, who shall be elected by ballot at the last meeting of each administration year and shall take office immediately at the close of the said meeting. A member who has served as Chairman for one year shall not be eligible to immediate re-election to the same office, *or to that of Vice-Chairman.* The Vice-Chairman shall succeed automatically to the Chairmanship in case of resignation or removal from Section territory of the Chairman.

The Governing Committee of the Section shall consist of the Chairman, the Vice-Chairman, the Secretary, the Treasurer, the Chairman who last held office, [and] the Chairman of the Standing Committee of the Section, *and the member of the Section representing it on the Sections Committee of the Society.*

[SB 9 Applications for Section membership shall be made on a form prescribed by the Governing Committee and shall be endorsed by the officers of the Section upon proof of eligibility.]

SB 10 The annual dues of members of the Section shall be $5 payable in advance at the beginning of the fiscal year. The [annual] dues of members elected during the fiscal year shall be payable on notice of election. *In case of election to the parent Society or of removal to Section territory from a non-Secton territory during the latter half of a fiscal year, one-half the amount of the annual dues shall be payable.* No enrollment of Section members *shall* [will] be made without payment of dues.

SB 11 A member who is dropped or expelled from *or whose resignation is accepted by* the Society shall [also be dropped from the membership of the Section upon receipt of notice of such action] *automatically cease to be a member of the Section.*

SB 13 An additional ticket bearing the signature of at least 12 members in good standing, *who are not candidates on said additional ticket,* may be filed with the Secretary not less than 20 days before the annual election. The ticket thus presented *by the additional nominating committee* shall be forwarded by the Secretary to all Section members in good standing not less than 10 days prior to the annual election, *and shall bear the names of the members of this additional nominating committee.* The canvass of the ballots cast for nominees of the additional nominating committee shall be in the same manner as provided for in the case of the regular nominees.

SB 18 The Treasurer of the Section shall pay out monies only upon vouchers duly authorized by the Governing Committee, which shall also act as an auditing committee for the accounts of the Section at the end of each fiscal year.

The Treasurer shall be placed under bond immediately upon his taking office, in an amount approximating the maximum of funds to be handled by him. The expense of such bonding shall be defrayed by the Society.

The reasons for the foregoing changes in the Section Constitution and By-Laws are discussed on p. 238 and an outline of the proposed changes in the Rules is given in connection therewith.

Present-Day Automotive-Fuel

By R. F. Lybeck[1]

New England Section Paper *Illustrated with* Diagrams

ABSTRACT

SKETCHING the rapid and substantial growth of the automotive and the petroleum industries, the author remarks the scientific laws that constitute definite limits beyond which automotive engineers and petroleum engineers cannot go to attain greater efficiency and discourses on the subject of motor-fuel quality. He states the present specification for the distillation range of gasoline, and mentions the conditions causing the transition from the former 400-deg. to the present 437-deg. fahr. end-point.

Since detonation is well established as being a function of the composition of the fuel, either something must be added to gasoline to cure this disease or detonation must be prevented by regulating the composition of gasoline so that the disturbing elements are absent, and the preventive method is the one followed by the company which the author represents.

By eliminating certain components of what is generally considered good gasoline—those in the "borderland" between kerosene and gasoline—the author's company finds that detonation can be eliminated and, to accomplish such elimination on a commercial scale, it developed new types of fractionation equipment consisting of a novel type of bubble-tower that permits very close "cutting" of the gasoline for the elimination of the "borderland" mixture, as well as a new process of treatment. The equipment built and installed worked as expected, and new ways to remove impurities were devised.

Tendencies toward and causes of corrosive effects of motor fuels are discussed, together with methods of testing for and of eliminating corrosive constituents.

FEW parallels to the rapid and substantial growth of the automotive and petroleum industries exist in industrial history. The automobile was still known as the "horseless carriage" 25 years ago, and gasoline was a by-product from oil refineries, having scarcely any economic value. Today this Country has about 20,000,000 automobiles, and it is estimated that they will have consumed about 10,000,000,000 gal. of gasoline during the year 1925. The automobile has thus become a distinct functional part of American life and business. It has become so important that any condition that might tend to impair its efficiency or stifle its growth would be nothing short of a national calamity.

The economic pressure brought to bear upon the petroleum industry is that it must continue to meet the ever increasing demand for fuel by producing every gallon of gasoline that can be made. No immediate danger of a gasoline famine is apparent, but the technical resources of the industry will be taxed continuously. Every effort to stimulate greater economy and efficiency should command the unqualified cooperation and support of all. Scientific laws constitute definite limitations beyond which automotive engineers cannot go to attain this greater efficiency. Limitations also exist beyond which petroleum refiners cannot go because of the nature of their products and other dominating influences. These limitations do not lessen our moral obligations to the American people as well as to ourselves; they mean that we must have a continuance of the spirit of cooperation now existing between the automotive industry and the petroleum industry.

The motorist expects and demands a gasoline that has for its characteristics easy starting, good pick-up and maximum mileage. Only a few years ago, specific gravity was the principal test for good gasoline and was directly an index of quality. When 76-deg. gasoline was available generally, specific gravity was the only quality test necessary; but the discovery of many new crude oils, the advent of the various blends and the general introduction of cracking processes, established the fact that there is no definite relationship between specific gravity and the really important characteristics of gasoline. For example, a gasoline made from a Pennsylvania crude oil will have a specific gravity of 58 to 60 deg. Baumé, and one made from California crude oil, will have a specific gravity of about 54 deg. Baumé. These two gasolines will have the same distillation range but, because of the inherent characteristics of the California crude oil, the gasoline made from it is a much better motor fuel. Gasoline made by the cracking process has also, generally, a lower specific gravity, but it has excellent volatility. Therefore, it is right that specific gravity is no longer included in our specifications for gasoline and such omission is a distinct advantage.

The specification for the color of gasoline is a distinct safeguard for the protection of the public against the unscrupulous or careless refiner. However, from a technical point of view, this cannot be considered seriously because the presence of a very small quantity of dark-colored oil will change the color of gasoline from white to brown without impairing any other physical characteristic. A gasoline made by the cracking process, although well within the defined limits on color, will often appear slightly off-color when compared with a water-white straight-run gasoline. This does not represent carelessness nor lack of refining on the part of the producer, but rather an inherent characteristic of the product, the entire elimination of which will be distinctly at the expense of the fuel and of the eventual consumer. Color and odor are the two properties of gasoline that are, perhaps, most apparent to the average motorist. Odor, fortunately, never has been included in a test. This would be impossible, for who can tell definitely what constitutes a good odor, a bad odor or a pleasing odor? It is entirely a personal matter. What is agreeable to one person will be offensive to another.

Specification for Gasoline

Regarding the question as to what gasoline is, many people have an idea that gasoline is a definite chemical compound; but it is not. Therefore, we have gasolines of the same distillation range and even of the same specific gravity that do not have the same characteristics when put to work in an engine. The most important specification is the distillation range and this is well

[1] Chemical engineer, Beacon Oil Co., Everett, Mass.

defined. The present specifications call for an initial boiling-point at 131 deg. fahr.; 20 per cent to be distilled off at 221 deg. fahr.; 50 per cent at 284 deg. fahr; 90 per cent at 392 deg. fahr. and a final boiling-point or end-point at 437 deg. fahr. The final boiling-point, or the point at which it is completely vaporized, is not higher than 437 deg. fahr.; and the foregoing are the specifications on which the Federal Government will buy gasoline for use as an ordinary motor-fuel. That type of gasoline will run an automobile, but it will not perform equally well in all parts of the Country.

The transition stage which has brought gasoline up to a 437-deg. fahr. end-point extended over the period from 1917 to 1923. Government specification was made for an end-point of 400 deg. fahr. in 1917, but today it is 437 deg. fahr. One can appreciate the amount of work that this change has forced upon the industry. A great amount of research work on the carbureter, on pre-heating, manifolding and hot-spotting was necessary in order that this fuel could be burned effectively. The carbureter gradually degenerated from its original purpose, that of a vaporizer, until it is only a metering device for the fuel and the air today. With this heavy gasoline came the attendant troubles of crankcase-oil dilution and excessive carbon-deposit due to the inability of all systems to handle such gasoline efficiently. All these things have caused much concern in the automotive and in the petroleum industries.

Detonation

Not only have these troubles appeared but the problem of "knocking," or detonation, exists. All are acquainted with the principal theories advanced to account for detonation. Thomas Midgley, Jr., T. A. Boyd and their co-workers in the General Motors Corporation Research Laboratories, and Dr. H. C. Dickinson, S. W. Sparrow and other scientists at the Bureau of Standards have contributed many data in the last 5 years, and the general fact is admitted almost universally that knocking is a function of the chemical composition of the fuel. Therefore, the regulation of gasoline knocking is strictly up to the refiner, who must use the crude oils which, because of inherent qualities, will yield better anti-knock gasolines, or else devise some other way to prevent knocking.

Curative medicine and preventive medicine are two divisions of medical science. The cause of the disease called knocking or detonation is well established as being a function of the composition of the fuel, and one way to cure the disease is to add something to the gasoline. That is similar to the curative method in medicine. The other way is to prevent the knocking by regulating the composition of the gasoline so that the disturbing elements are absent. That is similar to the preventive method in medicine. We attacked the problem with preventive medicine in mind. We find that, by eliminating certain components of what is generally considered good gasoline and which we have named the "borderland," we can eliminate knocking. To carry out the elimination of this objectionable part of the gasoline on a commercial scale meant the development of new types of fractionation equipment as well as new processes of treatment. This was done at our refinery. Equipment was built and installed, and it worked as was expected. We had to revise our ways of treatment to remove impurities. The new fractionation equipment consists of a novel type of bubble-tower that permits very close "cutting" of the gasoline for the elimination of "borderland" mixture.

Gasoline from "Cracked" Crude Oil

Gasoline made by the cracking process has always been considered superior to straight-run gasoline of the paraffin series as a motor fuel. This is true primarily because of the higher carbon-content and the lower hydrogen-content, due to the presence of aromatic and naphthenic compounds produced during the cracking cycle. The drawbacks of gasoline made by the cracking process, from the point of view of the refiner, have been color, odor and inability to treat it chemically without an abnormal loss. However, all these difficulties have been solved, and gasoline derived by the cracking process stands today as a most satisfactory motor-fuel. Harry Ricardo says:

> These fuels that have the higher volatility and stability always tend toward a greater return of the energy in them than those of low volatility and inferior stability.

Whether the distillation range represents the best method of measuring the volatility of gasoline exactly is a moot question. R. E. Wilson and others have challenged the test, but it stands at present as the accepted test. The distillation range affects two characteristics of gasoline; first, quick starting and acceleration as governed by the lower end of the curve and, second, distribution or complete vaporization and knocking as governed by the last part of the distillation range. With the many varying climatic conditions that prevail in this Country at the same season of the year, it is difficult to make definite specifications for fuel that will function efficiently in all localities. Situated as we are in New England, which has cold winters and hot summers, it has been our policy to market two types of gasoline, one for winter use and the other for summer use. This is a distinct advantage to the motorist, for it gives him satisfactory engine performance the year round.

Tests for Corrosive Effects

The test applied to gasoline to determine the presence of corrosive substances is that commonly known as the copper-strip test. A strip of brightly polished copper is immersed in the sample being tested for 3 hr., at a temperature of 122 deg. fahr. Corrosive sulphur-compounds are indicated if the copper becomes discolored due to the formation on the strip of copper sulphide. This test is of importance because of the general use of copper in the fuel systems of all automobiles. In addition to the test just mentioned for corrosive sulphur-compounds, a new specification for total sulphur was added a year ago which calls for the maximum of 0.1 per cent of sulphur. This test was added because of the apprehension in certain quarters that, although sulphur might not be present in corrosive form as indicated by the corrosion test, it might be of sufficient quantity to injure the working parts of engines. It has been the policy of some companies to combine benzol with gasoline. This may be a good thing in some instances and in other cases a bad thing. If benzol is not chemically treated, washed and neutralized, it then becomes destructive when added to gasoline. The officials of the General Motors Corporation discovered a troublesome motor-fuel condition in Baltimore, and in Birmingham, Ala., due to the use of a benzol that had not been refined properly and which, consequently, produced an acid condition in engines using the fuel. The acidity of a gasoline is generally due to the presence of mineral acids and should never be allowed under any circumstances. It is an indication of careless treatment on the part of the refiner. Thus, the relative values of different gaso-

FIG. 1—PRESSURE-TIME CURVE OF GASOLINE
Obtained with a Midgley Indicator from a Single-Cylinder Air-Cooled Engine To Determine the Detonation Value of Gasoline. When Detonation Occurs in the Cylinder, It Is Evidenced by the Sharp Peaks in the Curve

lines can be determined readily when the physical and the chemical tests, plus the engine performance, are applied.

THE DISCUSSION

QUESTION:—What is the scientific explanation of knocking?

R. F. LYBECK:—The explanation seems to be that it is a resonant wave due to abnormal pressure-conditions after dead-center. Figs. 1 and 2 depict pressure-time curves obtained with a Midgley indicator, and these show very clearly the knocking or detonation value of gasoline. Time and pressure constitute the coordinates. The data were obtained from a single-cylinder air-cooled engine having a normal compression-ratio, and under normal conditions of load, mixture, timing, and speed. When knocking occurs in the cylinder, it is evidenced by the sharp peaks in the curve shown in Fig. 1. A gasoline free from knocking, or detonation, produces a smooth curve such as is shown in Fig. 2.

M. R. WOLFARD[2]:—The best explanation of knocking is that abnormal pressure develops. Some gasolines will not produce knocking in a clean engine. If carbon accumulates and it is in the nature of a dry carbon, when the cylinder-head is removed the carbon will be found to have cracked so that minute points project which become hot points, and then knocking will occur. It is difficult to understand why it is that, with an engine interior covered with carbon points, there should be very marked knocking that perhaps can be reduced by installing new spark-plugs; if spark-plugs are put in that eliminate the pressure wave, the knocking stops. Every car driver is up against this problem of knocking. If a carbon deposit exists and the engine is operated on a lean mixture, knocking will occur on hills with ordinary gasoline unless the carbon is removed.

MR. LYBECK:—If a gasoline is used which has as an inherent virtue the elimination of knocking, the engine will run without knocking.

QUESTION:—Can one expect greater economy in mileage with such gasoline?

MR. LYBECK:—Yes; the efficiency is increased because no loss of power occurs on the power stroke due to knocking. With a gasoline which does not knock, the "heavy end," or so-called "borderland," is absent and, in that way, crankcase-oil dilution is decreased. The amount of priming needed is also reduced to the minimum.

[2] M.S.A.E.—Engineer in charge of research laboratory, Hopewell Bros., Watertown, Mass.

[3] Sales Department, Gulf Refining Co., Boston.

[4] President, Beacon Oil Co., Everett, Mass.

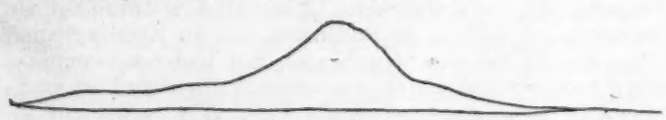

FIG. 2—PRESSURE-TIME CURVE OF GASOLINE
Obtained with a Midgley Indicator from a Single-Cylinder Air-Cooled Engine To Determine the Detonation Value of Gasoline. A Gasoline That Is Free from Detonation Produces a Smooth Curve Such as Is Shown

S. T. DOUGLAS[3]:—In connection with experiments made by the General Motors Corporation, the company I represent produces fuels that reduce knocking. The effort of the General Motors Corporation was to produce high-compression engines, and the average high-grade gasoline does not meet all requirements. Mr. Lybeck brought out the fact that the asphaltic crude oils produce cracked gasoline most readily. Has most of Mr. Lybeck's experimental work been done with asphaltic crude oils?

MR. LYBECK:—We have used crude oils having a naphthenic base mostly.

D. R. JUDSON:—If the new gasoline is substituted in an engine that is operated normally, has Mr. Lybeck any data showing the increase in mileage?

MR. LYBECK:—I have no data here that I can release; we are working on that problem now. Substitution of a better for a poorer gasoline would call for a readjustment of the carbureter to get maximum efficiency, although one would get better results with the better gasoline.

MR. WOLFARD:—Experiments made relating to the earlier designs of manifolds showed that opposite effects were obtained from those desired. The idea is to get uniform distribution in the cylinders. Some cars on the market have poor distribution, particularly when starting, because of the tendency to draw more gasoline into the end cylinders at this time. After many tests on engines having a small amount of carbon deposit, I obtained better mileage with them than with clean engines having no carbon deposit. The theory has been advanced that an engine should be built to produce knocking because that would increase the power from the fuel. I think that this is true, but, at the same time, it will knock an engine to pieces and no man wants to buy a new engine every week. You have to balance three or four factors, and it is rather amusing the forms these take.

V. A. NIELSEN:—The expression "borderland" is new to me. Just what is meant by the term and what is taken away when the "borderland" is removed?

H. L. WOLLENBERG[4]:—I coined the word "borderland." I think of the lighter end of crude oil as a family, a large one. The family group that includes gasoline and kerosene must be segregated and we must designate which members of the family constitute gasoline and which constitute kerosene. It is difficult to do this in that family, because there are so many children and relatives. If we draw a line, we would find some on one side and some on the other, so the term "borderland" is used to denote that part of the family group in which it is difficult to distinguish between the gasoline and the kerosene branches of the family. The custom has been, and still is, to crowd into the gasoline branch of the family as many as possible of the members that are on the "borderland" between gasoline and kerosene. An ordinary refining apparatus is incapable of straightening out these family relationships.

As to the idea of a "borderland," we created a *word* but not new *facts;* the term denotes the zone between gasoline and kerosene. We found that those members of the family who came from the "borderland" were the trouble makers in gasoline; they caused knocking, crankcase-oil dilution and sluggish engine-performance. When we keep the borderland members of the family out of gasoline, we put all the undesirable members of the family into kerosene. The gasoline that is free of borderland members is superior in volatility, anti-knock properties and for every feature of engine operation.

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Operating Costs and Economic Life of Motor Vehicles

By WILLIAM H. FAIRBANKS[1] LOS ANGELES GROUP PAPER

ABSTRACT

OPERATING-COST records, intelligently and painstakingly kept, are essential to the successful economical operation of a commercial fleet of motor vehicles. These records serve a two-fold purpose, first they provide the general management of the business with a reliable record of the total cost of the transportation service, from which can be deduced the unit cost per ton of merchandise, per gallon of gasoline or per passenger-mile, or whatever unit is applicable to the particular business involved, and second they afford the operating executives and engineers a current record of the itemized costs by classes of item and by individual vehicle and groups of vehicle segregated by types and sizes.

A table summarizing the operating costs of a large mixed fleet for 1 year shows how records of this kind reveal some highly important fundamental facts; for example, that the total annual operating-cost, exclusive of interest on investment, taxes, insurance, licenses, and drivers' wages, very nearly equals the total original investment in the vehicles; that this ratio is much higher for small low-priced vehicles than for large high-priced equipment; that the operating cost does not increase in proportion to the purchase price and the annual mileage; and that the ratio of variable operating-costs to fixed charges is less for the high-priced than for the low-priced equipment rendering comparable service mileage. Deductions from such facts are misleading, however, unless account is taken of other conditions not shown by the cost tables, such as differences in kind of service, in roads and in drivers.

While cost tables and depreciation curves provide a basis for estimating the economic life of trucks, no vehicle should be retired arbitrarily at the end of the period of economic life indicated by such guides. Unless a particular machine is in immediate need of a general overhauling at heavy expense, it may be advisable to continue it in service until the overhauling becomes imperative. If the cost of overhauling plus the salvage value is unduly high with relation to the purchase price of a new vehicle, the time is ripe to dispose of the old truck, but in the case of high-priced equipment that has rendered long service and whose salvage value is low, a comparatively heavy overhaul expense may be repaid by several years of additional economical service.

TO conduct any business successfully it is necessary to have exact records of the costs of carrying on the business. This is especially true as regards the operation of motor vehicles, in which line of work the operating-cost records serve a two-fold purpose by (*a*) showing how much is being spent to operate a fleet and (*b*) rendering it possible to examine the items of expense that compose the total cost with a view to effecting economies, stopping leaks and operating the vehicles more efficiently. Such an analysis cannot be made unless detailed costs are kept by individual cars and trucks and by groups of vehicles that are similar in type and size.

No matter how carefully and honestly the cost records are kept, however, the data must be studied and used with the greatest caution or they may lead to great confusion and to erroneous deductions. I do not believe that operating costs, for fleets at least, will prove the desirability or otherwise of any individual make of vehicle, because the comparative merits of different makes cannot be determined from operating costs except, possibly, those kept over a long period, since many factors, such as difference in operators, in road conditions and in service, cannot be shown in cost data and distort the significance of the figures.

PURPOSES SERVED BY OPERATING-COST DATA

As two distinct purposes are served by operating-cost data, two different kinds of records are kept, each kind being of value in its own field. The first set of records shows the total operating-cost of the fleet by the day, month or year and is of value to the executives of the company using the equipment, who can deduce from the record the unit cost of transportation. The over-all costs can be translated into unit costs according to the product handled; for example, many oil companies, I believe, reduce them to cost per gallon of gasoline and absorb the cost in the price of the fuel; other companies translate them into haulage cost per day per ton or per ton-mile or passenger-mile. Trucking companies measure these costs against income in corresponding units and fix their rates accordingly to determine whether they are making or losing money. These costs show the dollars spent to obtain certain results and are of particular interest to the executive charged with the general management.

The second set of records includes the minutia of separate items that enter into the total operating-cost. These records are of interest principally to the operating executives and engineers. They mean nothing and are only confusing unless interpreted by the special knowledge of the superintendent of transportation or the transportation engineer. Because the engineering, production and operation of motor vehicles are highly specialized technical professions, the figures showing the details of cost of operation are necessarily of greatest value and meaning to specialists who have been trained to analyze them.

FACTS REVEALED BY 1 YEAR'S RECORDS

The records of operating costs should be compiled according to the main essentials, as illustrated by Table 1, which is a random record of a recent year's operating costs of a mixed fleet of vehicles. This table covers all of the vital details of operating costs, I believe, with the exception of taxes, insurance, licenses, interest on investment, and drivers' wages. The first four of these excepted items are not carried in the motor-vehicle account of the organization from whose records the table was taken, and the last item, drivers' wages, is not applicable to the mechanical operation and maintenance of the vehicles.

It will be noted from this table that the average investment increases with the capacity and type of truck

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and that the ratio of fixed charges to variable costs increases in general in accordance with the initial investment; for example, in the Model-T Ford class the ratio of fixed to variable expenses is just about 1 to 1, whereas in the 2½-ton class the ratio is about 2 to 1.

Another interesting fact is the high ratio borne by the operating costs to the initial investment. Most persons, including many executives, think of the first cost of a motor vehicle as the main item of expense and are not familiar with the relatively high operating-expense. In the table the percentages of operating cost to investment in the several types of vehicle range from 36 to 160. Although not shown, the total annual operating-expense for the 306 vehicles amounts to 97 per cent of the total investment in them. Thus it is seen that it costs very nearly as much to operate this fleet for 1 year as the fleet cost to purchase, and this is exclusive of taxes, insurance, licenses, interest on investment, and drivers' wages.

Relation of Mileage to Operating Costs

The table also shows clearly the effect of mileage on operating costs. Apparently the 5-ton trucks cost less per mile to operate than the 2½-ton units, although the total operating-cost of the former for 1 year is 55 per cent of the investment, whereas, with the smaller trucks, the percentage is only 36. But the 5-ton trucks averaged nearly 100 per cent more mileage than the smaller ones, which so greatly reduced the fixed charges per mile as to bring the total cost per mile to less than 22 cents while that for the 2½-ton trucks was 26¼ cents, notwithstanding the fact that the variable costs were much higher for the larger units. At the same time, the capacity being double, the cost per ton-mile is only 42 per cent of the ton-mile cost of transportation with the 2½-ton trucks operated for only half the mileage.

On the other hand, the 2½-ton trucks, operated roughly about the same mileage as the Ford 1-ton trucks, cost nearly twice as much per mile to operate as the latter, but, having 2½ times the capacity, they were more economical to operate per ton-mile.

Records Essential to Successful Operation

The use of an accounting system that will currently give the costs both for individual cars and for car groups is absolutely necessary to the successful economical operation of a fleet, whether it be large or small. As "genius is the infinite capacity for taking pains," the secret of the successful commercial operation of motor vehicles lies in exercising extreme thoroughness in keeping and studying complete records of the various elements of operation, backed by specific knowledge of what one has accomplished and what he still has to attain. That is, one must know, first, the cost of the various items entering into operating expense, and, second, must constantly study the costs of these items individually to find wherein they can be reduced. For instance, the fuel cost for the heavy-duty equipment, as shown in the table, averages 4½ cents per mile and is 17 per cent of the total operating-expense for the 2½-ton trucks and 21 per cent for the 1-ton trucks. This phase at once presents itself as a promising field for investigation, not only by operators but also by the engineers at the engine and truck factories.

Repairs, depreciation, tires and garage service are also large items of expense to be watched constantly to prevent their insidious increase and, if possible, to reduce them.

I have dwelt at considerable length on operating costs because I believe the correct keeping and understanding of these is essential to successful and economical operation of commercial automotive equipment, whether such equipment is used by an individual or a company as a means of livelihood or by a company or corporation as a tool incidental to its operations. The importance of keeping itemized cost-records, as described, is fundamental and of universal application.

Determining Economic Life of a Truck

The economic life of a motor truck is an altogether different subject, and one on which, I believe, we do not have enough exact information. Some valuable informa-

TABLE 1—RECORD OF 1 YEAR'S OPERATING-COSTS OF A MIXED FLEET

Details of Fleet

	Model-T Ford	1-Ton Ford	1-Ton Gear-Shift Truck	2½-Ton Power-Equipped Truck	5-Ton Power-Equipped Truck
Average Number of Units Operated	170	88	30	12	6
Average Yearly Mileage per Car	10,600	6,350	8,500	6,550	13,750
Average Investment per Car	$500	$850	$1,600	$4,800	$5,500
Average Yearly Operating Expense	$800	$870	$1,300	$1,725	$3,000
Per Cent of Operating Expense to Investment	160	102	81	36	55

Operating Cost[1]

	Model-T Ford		1-Ton Ford		1-Ton Gear-Shift Truck		2½-Ton Power-Equipped Truck		5-Ton Power-Equipped Truck	
Variable	Cost per Mile	Relation to Total, Per Cent	Cost per Mile	Relation to Total, Per Cent	Cost per Mile	Relation to Total, Per Cent	Cost per Mile	Relation to Total, Per Cent	Cost per Mile	Relation to Total, Per Cent
Gasoline	$0.0108	14	$0.0160	12	$0.0167	11	$0.0448	17	$0.0459	21
Lubricants	0.0015	3	0.0025	2	0.0021	1	0.0032	1	0.0036	2
Tires and Tubes	0.0053	7	0.0088	7	0.0121	8	0.0015	1	0.0124	6
Repairs	0.0199	26	0.0329	24	0.0474	32	0.0383	14	0.0524	24
Total	$0.0375	50	$0.0602	45	$0.0783	52	$0.0878	33	$0.1143	53
Fixed										
Garage Service	$0.0246	32	$0.0424	31	$0.0320	21	$0.0398	15	$0.0193	9
Garage Rent	0.0040	5	0.0061	4	0.0046	3	0.0063	2	0.0031	1
Depreciation	0.0101	13	0.0273	20	0.0369	24	0.1285	50	0.0817	37
Total	$0.0387	50	$0.0758	55	$0.0735	48	$0.1746	67	$0.1041	47
Total Operating Cost per Mile	$0.0762		$0.1360		$0.1518		$0.2624		$0.2184	

[1] Exclusive of interest on investment, taxes, insurance, licenses and drivers' wages.

tion is furnished by the operating costs covering a period of time but such records are usually of too recent origin to be wholly conclusive. Moreover, so many changes have been made in truck design and materials of construction, and the growth of large truck-fleets has been so recent and rapid, that operating-cost figures extending back a number of years cannot safely be applied to present equipment and methods of maintenance. I have studied with keen interest many tables and curves by which attempts were made to show the economical life of a truck, but such methods appear to me to be like the life-expectation tables of the insurance companies, a measure of the average life of a number of units rather than an absolute measure of the economical serviceability of any individual truck.

For this reason, operating-cost tables and depreciation curves never should be applied arbitrarily and the economic utility of a truck decided by the time limit indicated by the curve. Most of such tables and curves indicate that the Model-T Ford should be disposed of after rendering about 25,000 miles, or 2½-years' service. This is sound theoretically but should not be followed in practice unless the actual physical condition of the unit indicates that such procedure is advisable. Although the table may show that after 25,000 miles the car should be traded-in, this should not be done unless the condition of the machine shows that it is necessary to spend a considerable sum for major repairs. If there is still 6 months' or 1 year's useful life in the machine without spending much money for repairs, by all means get that further service before disposing of the vehicle. I believe it to be good general practice to turn-in Fords at the end of 2½ or 3 years, medium-priced equipment at the end of about 5 years and high-priced equipment at the end of 7 years of service. In any case, however, I would not retire any vehicle until it is well past the sharp drop in value that occurs during the first year of service or before the depreciation curve has flattened out reasonably.

Relation of Overhaul Cost, Salvage Value and New-Truck Investment

The time for disposal of a unit should be decided by the relation of the amount that will have to be spent for repairs to its salvage value and its original cost. A 3-year-old unit that cost, new, $500, would cost at least $125 to overhaul completely. If the cost of overhauling is added to the salvage value, it will be found that the cost of the remaining life of the overhauled machine will be too high and that it will be unwise to spend the money for repairs. On the other hand, a 5-year-old heavy-duty truck that cost $5,500 can be overhauled at the maximum cost of $1,500, thereby adding probably 3 years of efficient life to the unit. The salvage value at the end of 5 years would be very low, the cost of replacing it with a new unit would be high and hence it would be wise to spend the money for the overhaul instead of trading-in the truck.

In every case, therefore, a study should be made of each unit after the normal average period of service, in either years or mileage, before deciding upon its retirement, instead of arbitrarily retiring all equipment in accordance with operating-cost tables and depreciation curves.

A SUGGESTED REMEDY FOR CRANKCASE OIL DILUTION

IN the paper with the above title by Robert E. Wilson and Robert E. Wilkin that appeared in the February issue of THE JOURNAL two unfortunate errors crept in during the preparation of the paper for publication. From the inscription on the upper curve in Fig. 4, on p. 165 it would appear that the "non-diluting" oil had picked up an additional 8.8 per cent of diluent in addition to the 10 per cent that was originally present. This is not in accordance with the facts. The 8.8 per cent in question and the 9.8-per cent dilution shown on the lower curve were the percentages of diluent present after 500 miles of running. The illustration in the correct form with the proper caption is printed herewith.

The other error occurred on p. 169 in connection with equation (10) near the bottom of the second column where a minus sign was omitted from the last parenthesis. This error is of course obvious from an inspection of equations (11) and (12). The equation should read as follows:

$$C = -(1/K_1)\log_e(X_e - X_p) \qquad (10)$$

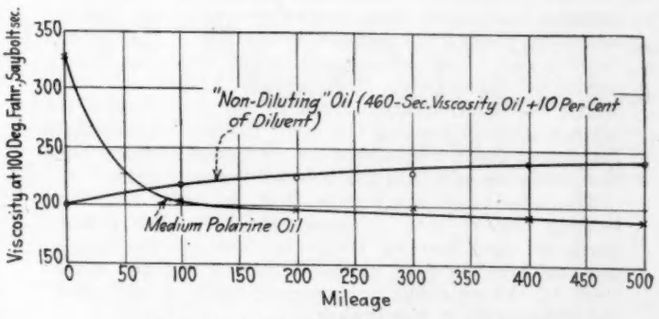

FIG. 4—VISCOSITY-MILEAGE CURVES

Average Viscosity Figures Obtained on over 90 Cars and Trucks Using Medium Polarine and "Non-Diluting" Oil Alternately under Summer Conditions. The Non-Diluting Oil Samples Showed an Average of 8.8 Per Cent Dilution after 500 Miles, as Compared with 9.8 Per Cent for the Medium Polarine Samples

Brakes for Heavy Vehicles

By L. C. Huck[1]

Cleveland Section Paper *Illustrated with* Drawings and Photographs

ABSTRACT

A HEAVY high-speed vehicle, such as a modern motorcoach, combines weight and speed, requires frequent brake-applications and necessitates the dissipation of an increased quantity of heat. As it weighs about four times as much as a passenger car approximately four times the braking-effort is required; consequently four times as much pedal-pressure must be exerted. To supplement the driver's muscular equipment, some outside force, such as compressed air, the vacuum of the intake-manifold or the inertia of the moving vehicle must be utilized. The author limits his present discussion to the use of the last-named force which he terms the use of "self-actuation," and also to its application to the rigid-shoe type of internal wheel-brake. Through mathematical analysis he determines the effect of self-actuation measured in terms of the increase or decrease of the cam pressure required to sustain the same normal pressure before and after an outside torque has been applied to the brake drum. The term "actuation" is used in preference to "energization," because the action of actuation is said to be precisely the same as the action of the cam pressure, namely, to rotate the shoe about its anchor-pin; and this action is usually spoken of as one of actuation rather than of energization.

After defining the terms "actuation factor" and "actuation constant," the author, by further mathematical analysis, determines their values under different conditions. Among other points considered are the determination of the limits of the safe arc of contact without the possibility of grabbing; the division of pressure between the shoes when journaled and when floating cams are used; a comparison of the life of two sets of floating-cam brakes of unequal length with that of conventional side-by-side floating-cam brakes of equal width, fitted into a brake drum of the same width; the method used in floating the cam; the use of aluminum for shoes; and a comparison of the braking-efforts with the journaled and with the floating types of cam.

The conclusion reached is that, in order that any braking-system may operate efficiently, the linkage must be rigid and its geometry must be correct, so that neither the road spring-deflection nor the movement of the axle due to torsional reaction will affect the adjustment of the brakes.

BRAKES for heavy vehicles is indeed a very serious and difficult problem. This is especially true in the case of heavy high-speed vehicles such as the modern motorcoach. Not only does a motorcoach combine weight and speed, but it also introduces the factor of more frequent brake-applications, with the attendant problem of dissipating an increased quantity of heat. In designing brakes, we must assume a certain coefficient of friction and, as this coefficient of friction is apt to vary with the temperature, it becomes apparent at once how important the problem of heat dissipation is.

Were we able to use the design of passenger-car brakes as a standard, the problem would not be difficult. Unfortunately, we cannot increase the diameter of the brake drum in proportion to the increase of weight. In the case of a heavy fully loaded motorcoach mounted on 34 x 7-in. dual tires, a mass of possibly 20,000 lb. must be stopped with very nearly the same brake-drum diameter that is used on a passenger car weighing less than 5000 lb.; in other words, about four times the braking-effort is required. The operator, consequently, must exert four times as much pedal-pressure. The effort required per unit of increase of pedal-pressure really seems to multiply as the pedal-pressure assumes greater proportions.

Supplementary Forces Assist Muscular Effort

As operators of heavy vehicles do not, as a rule, have muscular equipment proportional to the weight of their vehicles, an outside force must be brought into play to assist the muscular effort and to reduce the pedal-pressures to those necessary for braking a passenger car. A number of outside forces, however, can be used, such as compressed air, intake-manifold vacuum, inertia of the moving vehicle, and the like. This discussion will be limited to the use of the last-named force, which I have termed the use of "self-actuation."

Besides having greater ease of operation, brakes that are as sensitive as possible without being "grabby" are desirable, for they allow a reduction of the mechanical advantage in the brake-operating linkage. This reduction, in turn, results in less frequent adjustments for wear, elimination of the necessity for an uncomfortably long pedal-throw and, in the case of internal brakes, makes it possible to "follow-up" the expansion of the brake drums caused by heating.

The most undesirable features of a propeller-shaft brake are the increased wear and tear of the driving-mechanism between the wheels and the brake, and the fact that, for the same sensitiveness, the mechanical advantage of the linkage must be made inversely proportional to the axle reduction. The former disadvantage is of little importance if the brake is used only as a parking brake. Unfortunately, however, even though it were so labeled, the operator would use it as a foot or "service" brake if it were more effective than the hand or "emergency" brake. Making the operating-lever of this brake inaccessible might overcome this difficulty, but the brake would then be almost useless as an emergency brake. Accordingly, we shall consider wheel brakes only.

Rigid-Shoe Type of Internal Brake

The use of disc wheels, practically housing the entire braking-mechanism, is a factor which makes it desirable that we limit ourselves to an internal brake. This type of brake has the advantage of being more easily housed-in for the exclusion of dirt, water, dust, and the like, and allows better cooling of the brake drums. An analysis of the self-actuation of all types of internal brake would lead to an extremely long discussion; for this reason, I shall limit this paper to a discussion of the self-actuation of a rigid-shoe type of internal brake. Though the paper will deal with brakes for heavy vehicles, the principles are equally applicable to brakes for lighter vehicles and for passenger cars.

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Fig. 1 shows an elementary brakeshoe and brake drum. As this brakeshoe has only one point of contact, if we assume the brake drum to be free to rotate, we can write $Pa = Nb$.

If now the brake drum is rotated in a clockwise direction, a frictional force F acts on the brakeshoe through the lever-arm c, tending to rotate the shoe about its anchor-pin center in the same direction as does the force P. Then, $P'a + Fc = N'b$ and P and N denote the cam and normal pressures respectively with no outside torque on the brake drum, and P' and N' denote the same pressures when the brake drum is rotated by an outside force. If f is the coefficient of friction, $Fc = fN'c$ and we can write $P'a + fN'c = N'b$.

Effect of Application of Outside Torque

The effect, or self-actuation, when an outside torque is applied to the brake drum, may be measured in two ways, as follows:

(1) By finding the increase or decrease of the normal pressure N when the cam pressure P is held constant, that is, when $P = P'$; or

(2) By finding the increase or decrease of the cam pressure P when the normal pressure N is held constant, that is, when $N = N'$

In this discussion, method (2) will be followed and the effect, or self-actuation, will be measured in terms of the increase or decrease of the cam pressure required to sustain the same normal pressure before and after an outside torque has been applied to the brake drum.

It is obvious that if N' is taken to equal N, P' must be less than P, to balance the torques. Transposing, we can write,

$$P = (b/a)\,N$$
$$P' = (b - fc)\,N'/a$$

but, as the assumption has been made that $N = N'$, if we assume N and N' to be unity, we can write,

$$P = b/a$$
$$P' = (b - fc)/a$$

For counter-clockwise rotation, the direction of the frictional torque $fN'c$ would be reversed and the expression would be,

$$P' = (b + fc)/a$$

Thus, it is apparent that, for clockwise rotation, P' is less than P. For counter-clockwise rotation, P' would be greater than P, and, therefore, self-actuation due to clockwise rotation of the drum would respectively decrease and increase P for the upper shoe, as shown, and for an opposed shoe, which is not shown.

Self-Actuating and Non-Self-Actuating Shoes

A shoe in which the frictional torque is in the same direction as, or, in other words, assists, the cam torque has been termed a self-actuating shoe and, if in the opposite direction to the cam torque, a non-self-actuating shoe. The subscripts s and n will hereinafter be used to denote like functions of these shoes respectively, that is, P_s and P_n or P'_s and P'_n.

$P - P'$ expresses the difference between the pressure P required to give a normal pressure when the drum is stationary and the pressure P' required to give the same normal pressure with the drum rotating.

Substituting the values for P and P', we can write $P - P' = \pm f\,(c/a)$, + being for the self-actuating shoe and — being for the non-self-actuating shoe. $(P - P')/P$ expresses the ratio of the amount of decrease of the original force P; and, again, by substituting the values for P and P', we can write $(P - P')/P = \pm f(c/b)$, + being for the self-actuating shoe and — being for the non-self-actuating shoe. Then, $1 - [\pm f(c/b)]$ expresses the ratio of P' to P.

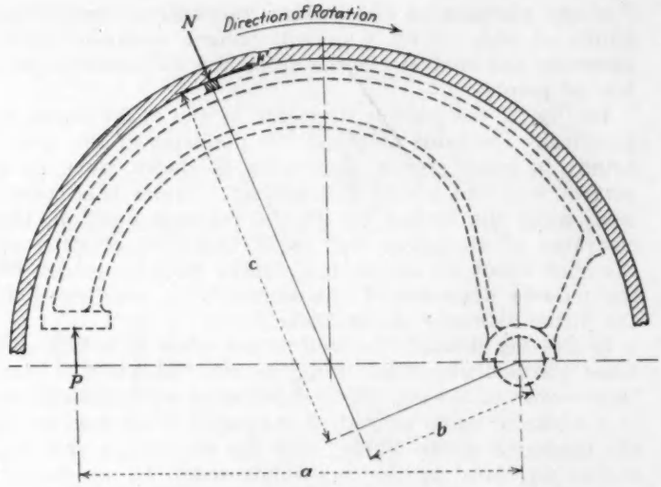

Fig. 1—Diagrammatic Sketch of Elementary Brakeshoe and Brake Drum

If the Brake Drum Is Rotated in a Clockwise Direction, a Frictional Force F Acts on the Brakeshoe through the Lever-Arm c and Tends To Rotate the Shoe about Its Anchor-Pin Center in the Same Direction as Does the Force P

The factor, $(P - P')/P$, when $N = N'$, has been termed the "actuation factor," and will be represented hereinafter by A. The factor A/f, designated hereinafter by Q, where f is the coefficient of friction, has been termed the "actuation constant," for it will appear later that Q is a constant for any given brakeshoe.

Thus, $A_s = (P - P')/P$, and $Q_s = (P - P')/Pf$ are functions of the self-actuating or upper shoe for clockwise drum-rotation, and $A_n = (P' - P)/P$, and $Q_n = (P' - P)/Pf$ are functions of the non-self-actuating shoe.

Actuation versus Energization

The term "actuation" is used in preference to the term "energization," for the action of the actuation is precisely the same as the action of the cam pressure, namely, to rotate the shoe about its anchor-pin; and we usually speak of the cam as actuating the shoe rather than energizing it.

Refer now to Fig. 2, which illustrates a conventional brakeshoe having lining contact from 25 to 159 deg.

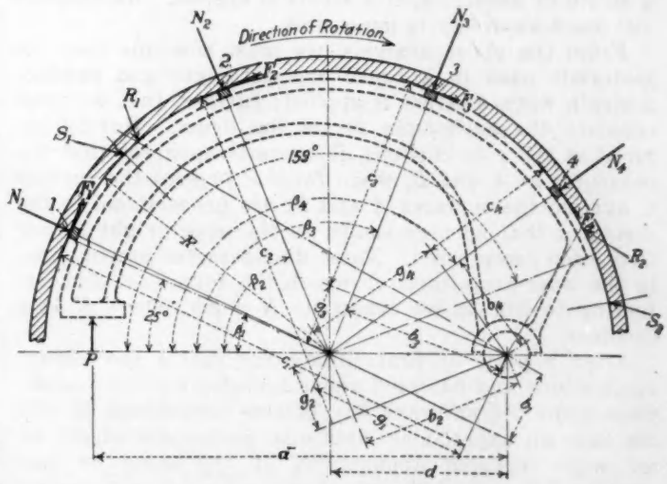

Fig. 2—A Conventional Brakeshoe Having Lining Contact from 25 to 159 Deg.

The Outside Diameter of the Brake-Lining Is Identically the Same as the Inside Diameter of the Brake Drum; in Other Words, the Lining Is Shown Worn to a Perfect Seat and Giving Full Contact

For the purpose of illustration we will use only four points on this lining. As will become apparent later, however, the analysis is the same for an infinite number of points.

In Fig. 2, the outside diameter of the brake-lining is identically the same as the inside diameter of the brake drum; in other words, the lining is shown worn to a perfect seat and giving full contact. Under this condition, when the brakes are in the released position, the clearance at all points will result from the rotation of the shoe about its anchor-pin, rather than by reason of the outside diameter of the lining being smaller than the inside diameter of the brake drum.

In passing it might be well to note that to obtain the same contact when the lining is new as when it has been worn to a seat, the brakeshoes must be machined to a diameter equal to that of the inside drum less twice the thickness of the lining; and the anchor-pin and the anchor-pin hole in the brakeshoe must be absolutely correctly located.

Refer now to Fig. 3, which shows a brake drum and a conventional-type shoe. We have taken an increment of arc, which may be considered a straight line and, for the purpose of illustration, have magnified it infinitely. Considering the enlarged part of the figure in the upper right corner as an actual mechanism and assuming that the points of contact A and B are machined with infinite accuracy so that the surface C of the brakeshoe will become parallel to and make contact with the two points at exactly the same time, the statement that pressure at A will equal the pressure at B, with which statement many seem to agree, requires that one of the following assumptions be made.

Necessary Assumptions

(1) That for values of D possible in practice, all the parts are absolutely rigid, that is, the modulus of elasticity of all parts of the mechanism is infinity, which is to say, that an infinite stress will produce zero strain

(2) That for parts producing a strain when a stress is applied, lever-arm D is infinitely long, which would cause surface C_1 to remain parallel to surfaces A_1 and B_1 as lever D is swung to compress springs bearing against the surfaces A_1 and B_1

We can certainly not make assumption (1), as any material that an engineer has at his disposal will produce a strain or deflection, if a stress is applied. Assumption (2) most assuredly is impossible.

From the above analysis, we must conclude that the materials used in a brake are not rigid and produce a strain when a stress is applied; further, that we must consider the anchor-pin inside the drum. Having arrived at these conclusions, it becomes apparent that the pressures at A and B, when force P presses the surface C against the surfaces A and B, are proportional to the distances that springs shown in the upper right corner have been compressed. These distances are proportional to the lever-arms through which the forces act and, referring to Fig. 2, we can write $N = Kb$, where K is a constant.

After a piece of brake-lining has had a few severe applications and has bled under high-temperature conditions until the saturant has become carbonized, it will not take an appreciable additional permanent strain or set with repeated applications of the same or less severity. It is doubtful whether a piece of brake-lining can be called elastic but, for that matter, a cast-iron liner is not elastic. Brake-lining, however, must act as an elastic material under the pressures to which it is subjected in a brake, for the normal motion or wear diagram of any shoe practically coincides with the actual wear of that shoe when worn out in service. What discrepancy exists can be accounted for by the fact that a brakeshoe is not of uniform rigidity. The toe of the shoe, having a greater deflection for a given pressure, deflects more than the part of the shoe nearer the heel and transfers its pressure toward the heel of the shoe.

Brake-Lining Wear Proportional to Work Done

Brake-lining manufacturers tell us that brake-lining wear is proportional to the work done by it, regardless of the speed and the pressure conditions, if the temperature is held constant. Work is proportional to the frictional force multiplied by the distance through which the frictional force acts. As the distance rubbed through during a given period of wear is the same for any part of a brakeshoe, it follows that the wear is proportional to the frictional force. Further, as the frictional force is proportional to the normal pressure, if the coefficient of friction is the same for all parts of the liner, we can say that the wear and the normal pressure are proportional. Wear and normal motion are, in this case, synonymous terms. In our discussion of Fig. 3, we concluded that, for an elastic material, the pressure is proportional to the normal motion. Accordingly, the wear or normal motion resulting from actual service, when taken with the statement that wear is proportional to work, indicates that brake-lining must act as an elastic material similar to the springs shown in the drawing at the upper right corner.

Geometrical Determination of Distance Moved by Point

In Fig. 4 is illustrated a geometrical proof that the distance moved by a point on a circle along a line drawn through the point and normal to the circle, when the latter is rotated through an increment of angle about a point other than its own center, is proportional to the length of a perpendicular erected from the center of rotation to the normal. In the drawing O is the center of a brake drum and T is a brakeshoe pivot-point. The shoe is rotated through an increment of angle about T so that point L moves to K. Being an increment of arc, LK is considered a straight line and is perpendicular to LT, and HL is the component of KL in the direction of the normal.

As triangles HKL and MLT are similar, HL is proportional to MT. It must now be noted that $HL = K(MT)$, where K is a constant, and $MT = OT \sin \beta$; therefore, $HL = K\ (OT) \sin \beta$. The distance OT is a constant and is the same as the distance d, Fig. 2, between the anchor-pin center and the brake-drum center. It follows, therefore, that the normal motion and, in turn, the normal pressure, which we have found to be proportional to the normal motion, are proportional to $\sin \beta$ at any point about the circumference. Therefore, again referring to Fig. 2, we can write that, for any pressure P, the drum being stationary,

$$N \text{ (at any point)} = Kb$$

Then,

$$N \text{ (at any point)} = Kd \sin \beta$$

It is interesting to note that the locus of the point of intersection of any normal and a line perpendicular to it drawn through the anchor-pin center, lines c and b respectively in Fig. 2, is a circle the radius of which equals one-half the distance from the center of the drum to the center of the anchor-pin.

Referring again to Fig. 2, we note that

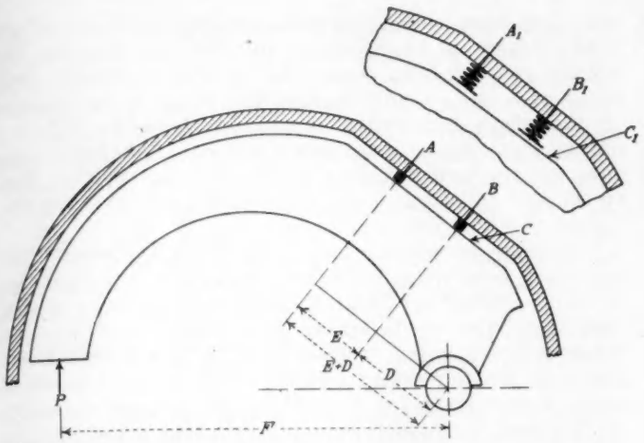

FIG. 3—VIEW OF BRAKE DRUM WITH CONVENTIONAL-TYPE SHOE

For Purposes of Illustration, an Increment of Arc, Which May Be Considered a Straight Line, Has Been Infinitely Magnified. The Modulus of Elasticity of All Parts of the Mechanism Is Infinity, That Is To Say, Infinite Stress Will Produce Zero Strain. In the Small Drawing in the Upper Right Corner the Assumption Is Made, That for All Parts Producing a Strain When Stress Is Applied, Lever-Arm D of the Main Drawing Is Infinitely Long, Which Would Cause the Surface C_1 To Remain Parallel to the Surfaces A_1 and B_1 as Lever D Is Swung To Compress the Springs Bearing on These Surfaces

$b = d \sin \beta$
$N = Kd \sin \beta$
$g = d \cos \beta$
$c = R \pm g$
$c = R + d \cos \beta$ ($\cos \beta$ having a negative value in the second quarter)

The actuation factor for any point is

$$A = (P - P')/P = \pm f\,(c/b),$$

and, by substitution

$$A = \pm f\,(R + d \cos \beta)/(d \sin \beta)$$

Now, instead of taking only one point of contact, let us take four points on the brakeshoe, namely, points 1, 2, 3, and 4, located by angles β_1, β_2, β_3, and β_4 respectively, and, by determining P and P' for the combined forces at points 1, 2, 3, and 4, determine the actuation factor for these combined points. It is interesting to note that the normal forces are distributed in a manner such that the points having the higher values of actuation have the smaller normal forces.

We can then write

$$P = (N_1b_1 + N_2b_2 + \ldots.)/a$$
$$P' = (N'_1b_1 + N'_2b_2 + \ldots.) \pm (fN'_1c_1 + fN'_2c_2 + \ldots.)/a$$

As, by definition of actuation factor, $N_1 = N'_1$ and $N_2 = N'_2$, and so on, we can write

$$A = (P - P')/P = \pm (fN_1c_1 + fN_2c_2 + \ldots.)/(N_1b_1 + N_2b_2 + \ldots.)$$

but, as the normal force at any point is proportional to the distance b for that point, we can write $N = Kb$, where K is a constant, and, therefore,

$$A = (P - P')/P = \pm fK\,(b_1c_1 + b_2c_2 + \ldots.)/K\,(b_1^2 + b_2^2 + \ldots.)$$
$$= \pm f\,(b_1c_1 + b_2c_2 + \ldots.)/(b_1^2 + b_2^2 + \ldots.)$$

To get the effective actuation-factor of a shoe accurately, a very large number of points would have to be taken, and this would require the measuring of a vast number of lines and considerable arithmetic. Accordingly, for absolutely accurate computations, we shall resort to calculus.

DETERMINATION OF POINTS BY CALCULUS

Let β_1 = the angle included between the toe of a shoe and a diameter drawn through the anchor-pin center.

Let β_2 = the angle included between the extreme toe-and-heel contact-points + the angle β_1.

Then

$$P = (WRKd^2 \int_{\beta_1}^{\beta_2} \sin^2 \beta d\beta)/a$$

where W = the width of the brakeshoe.

$$P' = WR\,[Kd^2 \int_{\beta_1}^{\beta_2} \sin^2 \beta d\beta \pm (fdKR \int_{\beta_1}^{\beta_2} \sin \beta d\beta + fd^2K \int_{\beta_1}^{\beta_2} \sin \beta \cos \beta d\beta)]/a$$

Since $A = (P - P')/P$, we have, by substitution,

$$A = \pm [(fR/d) \int_{\beta_1}^{\beta_2} \sin \beta d\beta + d/R \int_{\beta_1}^{\beta_2} \sin \beta \cos \beta d\beta]/\int_{\beta_1}^{\beta_2} \sin^2 \beta d\beta$$

For a shoe with equal arcs of contact on either side of a line drawn through the center of the shoe perpendicularly to a diameter through the anchor-pin center,

$$\int_{\beta_1}^{\beta_2} \sin \beta \cos \beta d\beta = 0$$

Therefore, for so symmetrical a shoe,

$$A = \pm [(fR/d) \int_{\beta_1}^{\beta_2} \sin \beta d\beta]/\int_{\beta_1}^{\beta_2} \sin^2 \beta d\beta$$

For the purpose of illustration, let us find A for the shoe shown in Fig. 2, when $\beta_1 = 25$ deg., $\beta_2 = 159$ deg., and $R = 1.6\,d$.

$$\int_{\beta_1}^{\beta_2} \sin \beta d\beta = \int_{25°}^{159°} \sin \beta d\beta$$
$$= -\cos \beta \Big]_{25°}^{159°}$$
$$= -[-0.9336 - 0.9063] = 1.8399$$

$$\int_{\beta_1}^{\beta_2} \sin \beta \cos \beta d\beta = 1/2 \int_{\beta_1}^{\beta_2} \sin 2\beta d\beta$$
$$= -1/4 \cos 2\beta \Big]_{25°}^{159°}$$
$$= -1/4\,(\cos 318° - \cos 50°)$$
$$= 1/4\,(0.7431 - 0.6428) = -0.0251$$

$$\int_{\beta_1}^{\beta_2} \sin^2 \beta d\beta = \int_{25°}^{159°} \sin^2 \beta d\beta$$
$$= [1/2\,\beta - 1/4 \sin 2\beta \Big]_{25°}^{159°}$$

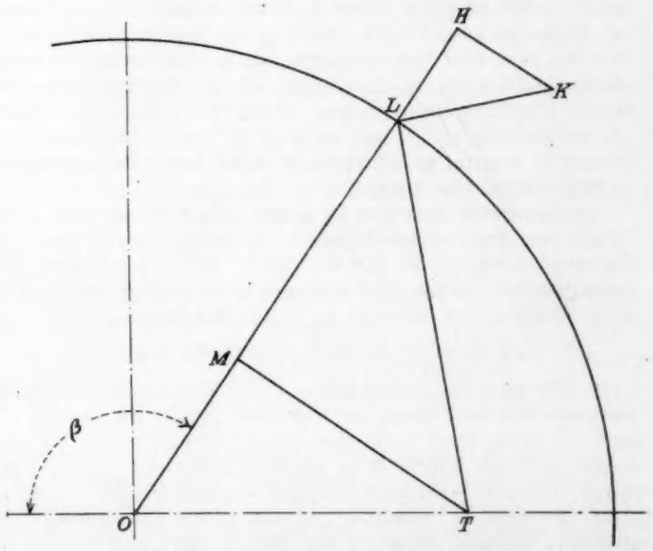

FIG. 4—GEOMETRICAL DEMONSTRATION

This Diagram Proves That the Distance Moved by a Point on a Circle along a Line Drawn through the Point and Normal to the Circle, when the Latter Is Rotated through an Increment of Angle about a Point Other Than Its Center, Is Proportional to the Length of a Perpendicular Erected from the Center of Rotation to the Normal. To Get the Effective Actuation-Factor a Large Number of Points Must be Determined, and Calculus Must be Used

$$= [1.3875 - 0.2181 + 1/4\ (0.6691) + 1/4\ (0.7660)]$$
$$= 1.5281$$
$$A = \pm f\ 1.6\ [1.8399 + 1.0/1.6\ (-0.0251)]/1.5281$$
$$= \pm 1.910\ f$$

Actuation of Point at Toe of Shoe

The actuation of a single point at the extreme toe of the shoe is $5.9f$. Therefore, Q, the actuation constant, is 5.9 and, as a point, will become self-locking when the coefficient of friction is greater than $1/Q$, which, in this case, is a coefficient of friction greater than 0.17.

We have also just computed A for the entire shoe to be $1.910f$. Therefore, if this shoe has full contact, it will lock when the coefficient of friction is greater than 0.524.

It should be noted that for a given actuation constant, Q, the actuation factor A varies directly as the coefficient of friction; also, that for a given length of lining and a given position thereof with respect to a diameter drawn through the anchor-pin center, the value of Q depends on R/d, which has been termed the anchor-pin ratio.

It has already been stated that any shoe will grab, if the coefficient of friction is greater than $1/Q$. Actually, it will not grab, if the coefficient is only very slightly higher, due to the springing-in of the toe of the shoe. It can be said that, within certain limits, the more flexible the shoe, the more the coefficient of friction for locking will exceed $1/Q$. It should be remembered that the actuation of any one point is resisted by the normal pressure at all other points as well, and this resistance is divided among the other points in proportion to the normal pressure at each of the points.

Various graphical methods have been suggested to determine the limits of the safe arc of contact, without the possibility of grabbing. Referring to Fig. 2, arc R_1R_2 and arc S_1S_2 have been determined by two such methods. The latter method is based on the coefficient of friction, and arc S_1S_2 has been determined for a coefficient of friction of 0.3. I have seen many hundred sets of brakes using the entire arc of contact shown in Fig. 2, namely, from 25 to 159 deg., and these brakes have never grabbed with coefficients of friction as high as 0.45, as determined on a Dodge brake-lining testing-machine, notwithstanding the fact that the graphical method indicated that the shoe would grab if the coefficient of friction exceeded 0.30. This can be explained by the fact that the theory on which this graphical method is based does not consider the normal pressure to be distributed in proportion to the sine of the angle β.

As has been observed in a preceding paragraph, if a brake has two opposed shoes, the actuation is positive for one and negative for the other; or it is positive and negative for the top and bottom shoes respectively in the case of clockwise rotation of the brake drum.

Journaled Versus Floating Cams

If the cam is journaled, it will divide its pressure between the two shoes in whatever proportion is necessary to give each shoe the wear corresponding to the angle through which it is rotated about its anchor-pin, when the cam is rotated through a given angle.

If the cam is "floating," it will always distribute its pressure equally between the two shoes; and the self-actuating shoe will always receive one-half the total cam-pressure. If the two shoes are symmetrical about a diameter drawn through the anchor-pin center, it is apparent at once that the self-actuating shoe for forward vehicle-motion will wear faster than the non-self-actuating shoe. As the greatest part of the cost of relining brakes is in laying-up the vehicle, removing the wheels and the like, it is good economy to reline both shoes of a set even though more lining is left on one of the shoes than on the other. Therefore, if, in a floating-cam brake, we shorten the non-self-actuating shoe for forward motion to make its wear equal to that of the self-actuating shoe, we shall not increase the frequency of brake-relining.

The exact proportioning of the shoes for equal wear under all conditions is impossible due to the variation of the coefficient of friction; but, even though the very lowest probable coefficient were assumed, in which case proper proportioning of the shoes for equal wear would make the non-self-actuating shoe longer than if the shoes were proportioned for the usual and higher coefficient of friction, the non-self-actuating shoe would still be shortened enough to enable another set of brakes to be put into the same drum, each set extending the entire width of the drum, and yet have the self-actuating shoe of the original set subtend as great an arc as could possibly be used without risk of grabbing. Figs. 5 and 6 illustrate such a set of brakes of 17.25-in. diameter. The shoes of the smaller set of brakes should also be of unequal length, the self-actuating shoe for rotation for forward vehicle-motion being made longer. Correct proportioning for equal wear is limited somewhat by the positions of the brake camshaft.

Comparison of Brakes

Comparing the life of the two sets of floating-cam brakes, shown in Figs. 5 and 6, with the life of conventional side-by-side floating-cam brakes of equal width, fitted into a brake drum of the same width, as used with the brakes shown in Figs. 5 and 6, it is obvious that the larger set of shoes that are used in the foot brakes has about the same life as both sets of side-by-side brakes would have, if used together, for the arc subtended by the self-actuating shoe of the larger set is about as long as can be used without a possibility of grabbing, and is as great as the side-by-side type of shoes would have. The smaller set of brakes, shown in Figs. 5 and 6, is operated as a hand brake and, although the self-actuating shoe for forward vehicle-motion is shorter than the side-by-side shoe would be, it has twice the width and consequently about the same life as one set of the side-by-side type operated individually. If both types of brake are operated in tandem, so that both sets of brakes will wear out simultaneously, the design illustrated in Figs. 5 and 6 will have about 50-per cent longer life than a design using conventional side-by-side brakes having the same total width.

Fig. 6 clearly illustrates the method used to float the cam. The inner ends of the camshafts are mounted in swivel-blocks. The release springs are very stiff and are hooked to a centralizer that is adjustable in length and can be clamped to the brake-spider in any position within the necessary limits. The adjustable feature of the release spring tension is not shown in Fig. 5. The clamping of these centralizers is accomplished by bolts, the nuts for which are on the exposed side of the brake-spider and are easily accessible. If the two brake shoes of a set are not proportioned so that the wear for the coefficient of friction of the particular lining being used will be exactly equal, the centralizer can be repositioned correctly by merely loosening and tightening again the centralizer bolts while the brakes are in the applied position. The centralizer block automatically takes the right position to centralize the shoes in the drum.

If, in a conventional type of journaled-cam brake, the cam is allowed to float by reason of clearance in the journal or the camshaft bushing, the brake will act as a floating-cam brake only until sufficient unequal wear has taken place to compensate for the clearance in the journal bushing. Furthermore, if this clearance is appreciable, the shoes will not be properly centered and, with a close brake-adjustment, the bottom shoe will drag, for the weight of both the shoes and the cam will be resting on the brake-drum, and the two shoes and the cam will rattle and bounce up and down, if the vehicle is traveling over rough roads.

Braking-Effort with Journaled and with Floating Cams

Let us now compare the braking-effect of any set of brakes, first, with a journaled type of cam and then with a floating cam, the actuating torque on the camshaft being equal in both cases; and let us see how this will vary the braking-action, or the effectiveness of a set of brakes.

Assume a conventional set of brakeshoes having a common anchor-pin with symmetrical shoes, and with an anchor-pin ratio and arcs of lining that will give an actuation factor A with the coefficient of friction of the lining used. Again, the subscript s will be used to indicate the self-actuating shoe, or shoe with positive actuation, and the subscript n to indicate the shoe with negative actuation. The "prime" indicates that the brake drum is rotating, and its absence, that the brake drum is stationary and has no outside torque acting upon it.

The braking action, or effectiveness, will be proportional to the summation of all the frictional forces on both shoes, which, in turn, will be proportional to the summation of the normal forces. In other words, $(\Sigma N'_s + \Sigma N'_n)$ for the floating-cam brake divided by $(\Sigma N'_s + \Sigma N'_n)$ for the journaled-cam brake will be the ratio of floating-cam to journaled-cam braking.

If T is the torque on the camshaft and r is the effective radius of each side of the cam, then T/r will equal the sum of the forces that the cam will deliver to the shoes. To simplify this example, we shall assume the lever-arm a, Fig. 2, through which the cam pressure acts, to be the same for both shoes. Release spring tension and frictional forces of the cam are negligible.

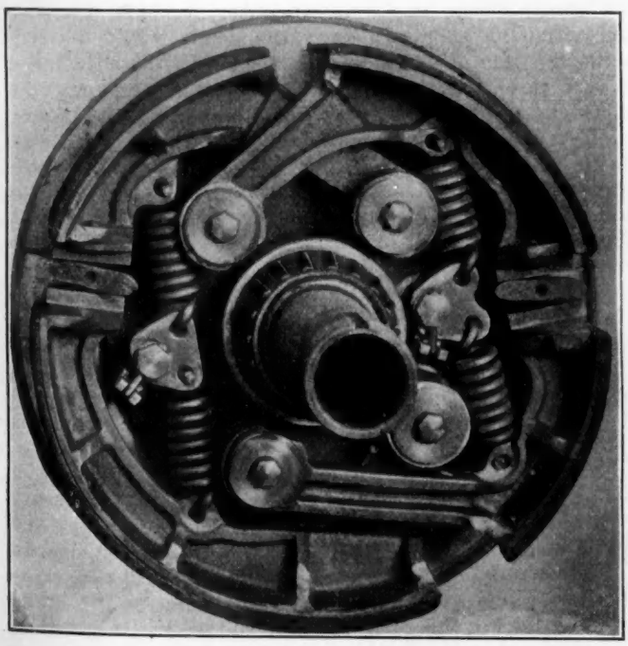

Fig. 5—End View of the Huck Double-Reduction Axle
This Axle Is Used for Motorcoaches Having Powerplants of 110 Hp. or less. The Spring-Pad Capacity is 14,000 Lb. and the Torque Capacity, 325 Lb-Ft.

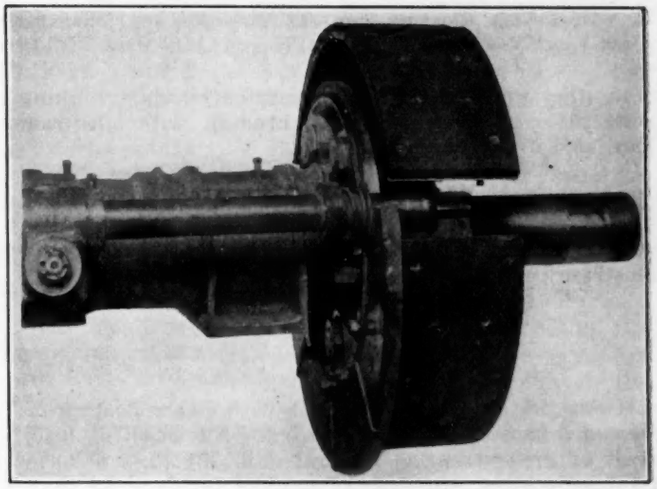

Fig. 6—Side View of the Double-Reduction Axle Shown in Fig. 5
The Non-Self-Actuating Shoes Are Shortened Enough To Enable Another Set of Brakes To Be Put into the Same Drum, Each Set Extending the Entire Length of the Drum, and Yet have the Self-Actuating Shoe of the Original Set Subtend as Great an Arc as Could Possibly Be Used without Risk of Grabbing

Cam Pressure

In the case of the floating cam, which acts as a differential, the cam pressure will always be divided equally between the two shoes. In the case of the journaled cam, the pressure will be divided between the shoes in whatever proportion is necessary to cause both shoes to wear equally. This fact is obvious when it is considered that, if one shoe wears faster than the other, the cam, being journaled, cannot follow the greater throw required to bring the faster-wearing shoe against the drum, and the latter will get no further cam-pressure until the other shoe has become worn sufficiently to allow the camshaft to rotate enough to let the cam bring the faster-wearing shoe against the drum again.

It has already been shown that wear is proportional to normal pressure. We can, therefore, say that, when rotation takes place, $\Sigma N'_s$ will equal $\Sigma N'_n$ in the case of the journaled cam. With the drum stationary, there is neither positive nor negative actuation on either of the shoes, whether the cam floats or not, and ΣN will represent the summation of the normal forces acting on each shoe in either case.

Now, let us find $\Sigma N'_s$ and $\Sigma N'_n$ for the shoes with the cam floating, in which case,

$$P'_s = P\,(1 - A) = T/2\,r\,(1 - A)$$
$$P'_n = P\,(1 + A) = T/2\,r\,(1 + A)$$

But, even though the brake drum is rotating,

$$P'_s = P'_n = T/2\,r$$

for the floating cam acts as a differential. Therefore, if a cam pressure of $T/2r$ results in ΣN_s and ΣN_n cam pressures of $T/2r\,(1 - A)$ and $T/2r\,(1 + A)$ will result in $\Sigma N'_s/(1 - A)$ and $\Sigma N'_n/(1 + A)$, respectively.

For the journaled cam, the pressures remain the same, and

$$\Sigma N_s = \Sigma N'_s,$$

also,

$$\Sigma N_n = \Sigma N'_n,$$

then,

$(\Sigma N'_s + \Sigma N'_n)$ (floating cam) $/ (\Sigma N'_s + \Sigma N'_n)$ (journaled cam) $= [\Sigma N'_s/(1-A) + \Sigma N'_n/(1+A)]/(\Sigma N'_s + \Sigma N'_n)$
$= 1/(1-A^2)$

In other words, for the same camshaft torque, braking with the cam floating equals braking with the cam journaled divided by $(1-A^2)$.

Value of A Floating Cam

The following numerical values for $1/(1-A^2)$ with various values of the actuation factor will more clearly illustrate the value of a floating cam:

If $A = 0.4$, then $1/(1-A^2) = 1.19$
If $A = 0.6$, then $1/(1-A^2) = 1.56$
If $A = 0.8$, then $1/(1-A^2) = 2.88$
If $A = 0.9$, then $1/(1-A^2) = 5.27$

If $A = 1.0$, the brakes will lock with either floating or journaled cams. With a coefficient of friction of 0.40, a set of brakeshoes, as illustrated in Fig. 2, will have a value for A of 0.76 and, by floating the cam, we shall multiply the braking action by 2.35.

It should be carefully noted that attempting to get results from floating cams by centering the shoes by frictional means, such as journaling the camshaft in an arm pivoted on the brake-spider, with frictional means to resist the rotation of the arm about its center rather than by resilient members that are thrown out of balance when the shoes are moved out of their correct center-position, does not accomplish the desired end. Such means will divide the cam pressure fairly equally between the two shoes, the amount of discrepancy depending on the unequal cam-pressure required to overcome the friction of the frictional device. This friction must be great enough to keep the shoes and the camshaft from "settling" as the unsprung parts pass over bumps or cobblestones. The sturdier the brake, the greater the frictional force that the friction device must produce, for, obviously, the brakes of a heavy vehicle must be stronger and, therefore, heavier.

The use of aluminum for shoes, to reduce their weight, is undesirable from the standpoint of rigidity, for the modulus of elasticity of aluminum is comparatively low and allows the shoe to deflect very much more than if the shoe were made of steel. The advantage of the greater heat-conductivity of aluminum is largely lost, due to the fact that the brake-liner acts as an insulator. This disadvantage does not, of course, apply to aluminum shoes with metal liners.

Importance of Rigid Shoes

Under severe emergency applications of the brake at high speeds, in which case the wheels are often suddenly locked or "dragged," the braking action required to overcome the inertia or momentum of the rotating wheels may be greater than the action required to overcome the frictional forces of the ground on the wheels. Consequently, it should be noted that to avoid any possibility of the shoes' taking a permanent set, they must be made more rigid than the vehicle weight alone would seem to indicate.

It must be remembered that, when the shoes of a set of brakes are properly centered and the camshaft is rotated, both shoes will strike the drum at the same instant. This holds true for either type of brake, namely, the floating-cam or the journaled-cam brake. In the case of a floating-cam brake, however, the camshaft, simultaneously with the application of an outside torque on the brake drum, will move toward the self-actuating shoe. This can be explained by the fact that, due to self-actuation, this shoe will have a greater normal-pressure than will the non-self-actuating shoe and, consequently, will compress its lining more and deflect more. If, under these conditions, the camshaft does not move toward the self-actuating shoe, it does not act as a floating cam, and the braking effort will not be increased by reason of self-actuation. In other words, if, in place of the frictional device, the arm in which the camshaft is journaled is rigidly clamped, or the friction of the frictional device is increased to a point at which it acts as a rigid clamp, we shall not benefit by self-actuation. The only advantage to be gained will be that the shoes can be centered, notwithstanding a variation in the thickness of the brake-lining and in other commercial tolerances.

Again, considering the frictional device, let us assume it to be designed for use on a motorcoach. In such a case the brakes would have to be of a very sturdy and heavy design, and the frictional torque required to swing the camshaft supporting-arm would have to be correspondingly great to avoid any possibility of the shoes' "settling." If, now, the brakes are applied with no torque on the brake drum, and one shoe were to strike the drum first, the shoes would center themselves in the drum as soon as enough pedal-pressure has been exerted to overcome the frictional torque of the device supporting both the right and the left camshafts.

Movement of Camshaft Toward Self-Actuating Shoe

If a torque is applied to the wheel and transmitted by it to the brake drum and if the frictional torque of the supporting-arm is not great enough to give the same result as if the arm were clamped rigidly to the spider, the camshaft will move toward the self-actuating shoe. This movement is due to the fact that the non-self-actuating shoe will have far less normal pressure than the self-actuating shoe, the pressures being respectively proportional to $1/(1+A)$ and $1/(1-A)$. Accordingly, the liner on the non-self-actuating shoe and the shoe itself will not give or deflect so much as the self-actuating shoe. When the brakes are then released with the camshaft in this new off-center position, the non-self-actuating shoe will clear the drum, while the self-actuating shoe is still under pressure. If the adjustment is very close, that is, if the total clearance between both shoes and the drum is very little, the self-actuating shoe may not clear at all, the non-self-actuating shoe having itself taken the total clearance available. The self-actuating shoe would then drag with a pressure that will depend on the distance that the camshaft had moved. This pressure could, of course, not be greater than the pressure required to move the camshaft supporting-arm against its frictional torque. The possibility of the brakes' dragging, as a result of this camshaft movement, will depend on the total clearance available, the severity of the brake application, the rigidity of the brakeshoe, and the compressibility of the brake-liner. A physical proof that this exact action takes place can be made as follows:

Take any internal brake using floating actuation means, and first apply the brake with great pressure with the vehicle stationary and with no torque on the brake drums. Then, without varying the pressure on the brake pedal, by racing the engine and with the transmission in low gear, apply a large torque to the brake drum by letting the clutch engage suddenly. It will then be noticed that the brake pedal has, without additional application of pressure, moved toward the floor-boards and will, without increasing the pedal pressure, stay in this new position when the brake drums have been re-

lieved of all torque. The motion of the pedal between the original and the new positions represents the greater deflection of the self-actuating shoe, and also the greater compression of the self-actuating shoe-liner. As has been set forth before, this is due to the greater normal-pressure, in other words, the greater braking-action, caused by self-actuation. If the brake is released and again applied with the original pressure, the brake pedal will return to the original position, unless the shoe, lining or drum has taken a permanent set.

Centering of Shoes

If the shoes are centered by the method illustrated in Fig. 5, the shoes will again become centered when the pedal pressure is released. As has been pointed out before, if a frictional device is used, it will be noted that the camshaft has remained in the new position, holding the shoes off-center, unless, in this new position, the normal forces are too great to be overcome by the frictional torque of the camshaft supporting-arm. Should this latter situation be the case, the shoes will move toward their correct centered position only far enough to allow the normal pressure and the torsional resistance of the camshaft supporting-arm to come to a balance. The brakes would then continue to drag, in an amount equal to the braking-action caused by this normal pressure, which, for forward direction of rotation, would most certainly not be negligible, as this shoe would then have positive actuation and be self-intensifying. To measure the amount of camshaft movement in a frictionally centered device, the test should be repeated with the drum rotating in the reverse direction, in which event the cam would move in the opposite direction, and the total camshaft movement could be measured.

The same action will, of course, but only to a very limited extent, take place in a so-called rigidly journaled cam-brake, due to the give or deflection of the camshaft support and the overhanging part of the shaft. As has been pointed out before, if the clearance in the camshaft bushing is appreciable, the shoe and the camshaft will rattle up and down to the extent of the clearance in the bushing.

Rigidity of Springs Essential

In the case of the "spring-balance" means of obtaining centralization, as illustrated in Fig. 5, the springs must be very rigid to overcome any appreciable movement of the shoes when the vehicle is passing over rough roads, for an appreciable shoe movement might, in the case of a close brake-adjustment, cause one or the other of the shoes to drag momentarily.

It has been found in practice, however, that the elimination of shoe movement is not the determining factor in computing the required stiffness of the release springs, for an even greater stiffness is necessary to avoid brakage due to vibratory movements of the shoes. In the brakes illustrated in Fig. 5, a stiffness of about 1800 lb. per in. deflection will avoid any possibility of brakage. It should be noted that the force tending to center the shoes is not proportional to the release spring tension but is entirely dependent on the stiffness, or pull, required to stretch the springs a unit distance. Thus, the springs will function equally well as a centering device, whether the pull exerted by them is 50 or 500 lb. With springs of 1800 lb. per in. deflection, a movement of 1/64 in. at the point where the springs are anchored would require a force of 56 lb. at that point. Any tendency to center incorrectly, either by reason of the weight of the shoes or the difference in the length of the lever-arm through which the cam works, distance a, in Figs. 1 and 2, can be corrected either by moving the release spring-hooks, or by mounting the cam plates at an angle.

In my opinion the shoes of brakes used on motor-coaches and other heavy vehicles should be made of forgings, to provide great rigidity. It will be noted that the design illustrated in Fig. 5 lends itself very well to such specifications.

Actuation Factor of Floating-Cam Brakes

In a floating-cam brake, it is important to design the self-actuating and non-self-actuating shoes so that they will have the same actuation factor, even though the shoes are not symmetrical, so that the braking actions in the forward and the reverse directions of vehicle motion will be equal.

The actuating factor must not be great enough to cause the brakes to grab, with the highest coefficients of friction found in standard linings, although the extremely high coefficients found in so-called "freak" brake-linings should, in my opinion, be disregarded, for, if the brakes were designed to avoid grabbing with such "trick" linings, they would not be so sensitive as is desirable with the standard makes of lining.

For the efficient operation of any braking-system, the linkage must be rigid and its geometry must be correct, so that neither the road spring-deflection nor the movement of the axle due to torsional reaction will affect the adjustment of the brakes.

THE DISCUSSION

C. A. Schell[2]:—After noting the enormous amount of work that brake designers have done in developing an efficient brake for large and high-speed motorcoaches, I am wondering just how long it would take them to increase the brake-drum diameter to the 21-in. standard, if it were adopted.

L. C. Huck:—A 17¼-in. brake, which is about the largest size that can be used on a 20-in. wheel and still have room for a few ribs to stiffen the drum, can be designed to stop the heaviest motorcoach easily, but the great trouble is caused by the heat generated.

In some of our experiments, we have set fire to the lining, the temper has even been drawn from the metal, and the grease has been distilled out of the hub. It seems as if the heat cannot get out. The heat does not seem to travel through the inside disc to the tire, for, if that were the case, the tire would also heat up. Most of the blow-outs have occurred on the outer rim of the inside wheel. It seems that the heat is transferred through a space filled with stagnant air and some of the heat gets to the rim by radiation. If the air can be caused to rotate with the wheel, and it does to a certain extent, the action would be the same as that of a centrifugal pump.

The brake drum covers the valve-ports, and most of the air thrown out around the tire by centrifugal action is drawn out through the ports. We intend to close those ports so that all the air must be drawn in past the brake drum; a cool cushion of air will be kept between the brake drum and the rim, and will serve to carry off by radiation whatever heat there is.

I pasted some paper over the outer ports of several wheels and, with no propelling action, causing the air between the tires and discs to rotate, I found a very remarkable decrease in the temperature under the outer rim of the inner tire. I cannot give the exact tempera-

[2] M.S.A.E.—Chief engineer, Thermoid Rubber Co., Cleveland.

ture, but I believe it was from 350 to 170 deg. fahr. If we can get that air between the discs, why can we not place a propeller or paddle-wheel there and draw in a current of air? The propeller could be made of rubber so that if both tires should go flat, it would not injure the tires.

H. D. HUKILL[3]:—When we began our experiments on power brakes 4 years ago, we proved the truth of some of the calculations that Mr. Huck has shown. We had an axle that was originally designed to take an 18-in. internal-expanding brake. In an effort to provide greater lining-area, we put 21-in. drums on that axle, built special 21-in. shoes, such as Mr. Huck has shown, and mounted them on an 18-in. anchorage. That gave very fine braking action, under normal conditions, with a coefficient of lining friction of about 0.40; but if the car was operated in rain or through slush, the coefficient of friction was reduced and the brakes were very ineffective. I think that bears out what Mr. Huck has said.

It was found that, with a coefficient of friction of about 0.55, this particular brake was self-locking, and the only way to release it was to back up the coach. The self-locking condition would be evident occasionally after the brakes had been used on a heavy grade, then allowed to cool off, when the binder that had been "cooked" out of the lining apparently caused high friction between the lining and the drum. We found that the coefficient of friction of fabric linings varies greatly under different conditions of temperature or moisture, and the results of our experiments led to the consideration of metal linings. For power-operated brakes, the most satisfactory brake is one in which the coefficient of friction is as uniform as possible.

Regardless of the temperature, the coefficient of friction between any two materials varies inversely with the speed. The higher the speed, the lower the coefficient of friction, even in the case of cast iron in contact with steel. Because of this, self-actuation is not desirable in power-operated brakes, and the most uniform braking-action is secured when the actuation factor is as low as possible.

Among the other advantages of metal lining is the fact that it will retain its coefficient of friction regardless of operating conditions. The brake is just as effective in wet weather as in dry weather. If the heat is dissipated and excessive tire temperatures are prevented, metal-to-metal brakes can be used on mountain grades, where fabric linings would burn out.

The tests made indicate that, with metal lining, the cooling is somewhat better than with fabric lining, one of the reasons being that, with the metal brake, the dust shield is removed. That, of course, has both advantages and disadvantages. It allows the brake to pick up considerable road dust and dirt, but the brake runs cool. Sixty air-operated metal-brake equipments are used on the Cleveland Railway Co.'s coaches, and their performance is being closely watched for further data on this class of service.

O. M. BURKHARDT[4]:—Mr. Huck's paper constitutes a very valuable contribution to the engineering literature on brakes. The development of an actuation-factor is distinctly new and should be of great practical value, especially as a means of comparing the self-energizing value of a contemplated design with one of known performance.

[3] M.S.A.E.—In charge of automotive division, Westinghouse Air Brake Co., Wilmerding, Pa.

[4] M.S.A.E.—Research manager, Society of Automotive Engineers, Inc., New York City.

[5] See THE JOURNAL, July 1925, p. 64.

Mr. Huck has endeavored to give a clear understanding of the need of more powerful brakes. Drum diameters can certainly not be increased to compensate for increases in speed and weight. Tire-makers are now seriously troubled with motorcoach tire and tube failures due to excessive heat, because the brake-drum diameter is very nearly equal to that of the inside diameter of the tire. This condition imposes a definite limitation on the brake-drum diameter. To obtain effective brakes, therefore, designers must look for some mechanical advantage. This obviously had led to a careful study of the wrapping action of brake members.

Mr. Huck makes it clear that his paper deals with brakes for heavy vehicles. He is the designer of an axle that was conceived and built for use in motor trucks. From this fact, we can safely conclude that he is thoroughly accustomed to thinking in terms of truck brakes.

It is well known that brakeshoes and brake drums, as used in motor trucks, are of much sturdier design than those used in much lighter passenger-cars. Because of lack of rigidity, certain difficulties that Mr. Huck has not discussed are encountered with passenger-car brakes. He states that he has seen many sets of brakes in which a much larger arc of contact was used than that determined by the graphical method set forth in my paper, Fundamentals of Brake Design[5], and that these brakes have not grabbed; but he does not say that they did not squeak or chatter.

The paper referred to was written to give a criterion of brake squeaking and chattering and, through a clear analysis, these phenomena were traced to lack of rigidity. Mr. Huck, however, did not discuss possible distortion in either the brake drum or the brakeshoe. His conclusions are based on certain assumptions that cannot be easily reconciled; and I believe that a further discussion of these assumptions would be beneficial and enlightening. It is, of course, obvious that some assumptions must be made in any complex treatise, if a practical solution is desired.

Mr. Huck's analysis hinges entirely on the supposition that the normal pressure N is directly proportional to the distance b and to a constant k. The distance b is mathematically expressed by the equation $b = d \sin \beta$; hence, he gives $N = kd \sin \beta$. This assumption would be tolerable if there were such a thing as a perfectly elastic liner, yielding much more readily than its environments, namely, the shoe and the drum. This, however, is far from being the case. Instead, the elastic compression of the liner under the influence of the customary specific pressures is exceedingly small as compared to the deflection of the shoe and the drum. It was shown in my paper that, for cast-iron liners this elastic compression will be less than 0.000001 in. It is obvious that brakeshoes and brake drums, under ordinary conditions, deflect thousands of times this amount; consequently, this supposition can be universally applied only to brakes so designed that nothing except the liner will deflect under load. This is equivalent to discussing the deflection of a leaf-spring in terms of the diminution taking place within the solid height of the leaves rather than considering the deflection of the leaves themselves.

If pressure from the brakeshoe is exerted on the drum, the drum will become out-of-round and will form an ellipse with the major axis in the direction in which the force is applied and reacted. The forces acting on the brakeshoes, moreover, will change their curvature from that of a concentric circle to that of an ellipse with its major axis perpendicular to the direction of the applied forces. The difficulty, therefore, arises from the fact

that the brakeshoes and the brake drum form two ellipses with their major axes perpendicular to each other. Because of this condition, contact is likely to take place at the ends of the shoe where the normal pressures are likely to increase beyond control, if the arc of contact is not held within definite limits. What will happen under adverse conditions has been clearly stated in my paper; it can be said that squeaking and chattering brakes have been cured by holding the arc of contact within the limits determined by my graphical method.

MR. HUCK:—The points Mr. Burkhardt has brought up in his discussion are indeed well taken. I believe, however, that, although his criticisms of my paper are entirely in place, they are justified more by reason of certain omissions in the paper, principally omissions of mathematical deductions, than because of any fallacies in the reasoning. The following supplementary explanation will serve to show that brake-drum and brakeshoe distortions have been taken into consideration in the derivation of the formulas, though I am frank to admit no direct reference was made to them in the paper, principally for the sake of brevity.

Referring to Fig. 4, it was shown that, if a free brakeshoe is rotated about its anchor-pin, the radial motion LH at any point on the shoe is proportional to MT. The proportionality factor is LK/LT, which is the tangent of the angle LTK. As the angle through which a shoe can be rotated about its anchor-pin by the usual brake-linkage and pedal-throw, after all the clearance has been taken up, is comparatively small, the angle LTK in radians can be used in place of the tangent of the angle LTK. It is obvious that all points on a free shoe will rotate about the anchor-pin center through this same angle. Referring now to Figs. 1 and 2, and indicating the angle LTK through which the shoe swings by α, in radians, we can write: The radial motion at any point equals α times the lever-arm b for that point and, if the point is located by an angle β measured from a diameter drawn through the anchor-pin, we can write: The radial motion $= \alpha d \sin \beta$. We are, however, interested in radial or normal pressure rather than radial or normal motion and it therefore becomes necessary to write an expression for the pressure resulting from a given amount of normal motion. Let us assume three springs of various rates, No. 1 to have a rate of A lb. per in., No. 2, B lb. per in., and No. 3, C lb. per in. If these three springs are now used together, or in series, their combined rate will be the reciprocal of the sum of the reciprocals of the three rates, that is, $1/(1/A + 1/B + 1/C)$, which expression we will term S. In this analogy, the drum, liner and shoe are represented by the three springs of different rates. As the brake-lining is not a perfectly elastic material, and as the stress-strain relation of the brakeshoe proper is not the same for all points on the brakeshoe, the analogy is not exactly true, but I believe is nearly accurate enough for practical purposes, as has been proved by the fact that the actual and the theoretical results check very closely. The possibility of error will be more completely discussed in a later paragraph.

If an outward stress or pressure is applied at a point on the brake drum, a given deflection will result, the amount of which will depend on the stress-strain relation for that point, and this deflection will be proportional to the stress, unless, of course, the materials are stressed beyond the elastic-limit. If the brake drum is a perfect circle, this stress-strain relation will be the same for any point around the circumference at the intersection of a plane perpendicular to the axis of the brake drum and the rim of the brake drum, as the section resisting the pressure is symmetrical at any point about the circumference. As Mr. Burkhardt has brought out, this is not entirely true if the drum becomes an ellipse but, for the drum distortions allowable, it is so nearly true that I believe the error is negligible; for, if a brake drum is forced out-of-round more than 1/16 in., it is too light for the pressures to which it is being subjected, and a difference of ⅛ in. in the major and the minor axes on a mean diameter of, let us say, 14 to 18 in. would not introduce a serious error. At least, we surely are safe in saying that the greater the deflection, the greater must be the pressure producing it.

I must differ with Mr. Burkhardt in his statement that the brakeshoe and the brake drum actually form two ellipses with their major axes perpendicular to each other. Even if both the drum and the shoes deflect, the distorted outside diameter of the shoes must coincide with the distorted inside diameter of the drum and, if the distortion takes the shape of an ellipse, the major axes of the drum and the shoes must coincide. This is obvious for, if the major axes of the two elements were at right-angles to each other, there would be no contact at the part of the brake drum adjacent to the ends of its major axis, and consequently there would be no forces on those parts of the drum to distort it in this manner.

The lining is not truly an elastic material, unless a metal liner is considered. In the case of a fabric lining, within the limits of the specific pressure used and at temperatures below 300 deg. fahr., the stress-strain relation increases as the stress increases. In other words, at higher stresses a given deflection will require a greater increase in stress than at lower stresses. We are, however, surely safe in saying that the greater the deflection, or, in this case the compression, the greater must be the pressure producing it.

The largest error creeps in when considering the brakeshoe. This is really a beam and the stress-strain relation increases as we move away from the toe, or cam end of the shoe. That is to say, a given radial deflection will require less stress or pressure at the toe than at the heel. To write an expression for this variation and carry it through the integration and computations for the actuation-factor would involve some deep mathematics and I believe it doubtful whether, even if such an expression could be carried through as a variable, the greater resulting accuracy would have any practical value. Again, in the case of the brakeshoe, we are certainly safe in saying that the greater the deflection, the greater must be the pressure producing it.

It is rather confusing to consider the deflection as the independent variable and the stress or pressure as the dependent variable, as this relation is usually spoken of conversely. In the case of a brakeshoe, liner and drum, however, the former sequence is the logical way of considering the problem, for parts deflect against elastic, or at least part elastic, resistance and an endeavor is made to find the resulting stress or pressure.

Assuming elastic characteristics for the three elements involved and further assuming the shoe to have a uniform stress-strain relation, the constant S, as derived in the analogy of the three springs of different rates, would represent the stress-strain relation at any point of the combination of the drum, liner and shoe used simultaneously, and cognizance of both shoe and drum distortion would accordingly be taken. Now then, as $\alpha d \sin \beta$ represents the radial motion, which in this case is the deflection, $S \alpha d \sin \beta$ will represent the stress or

pressure at any point. In my paper, the factor $S\alpha$ was termed K. The discrepancy due to the non-uniformity of the stress-strain relation of the brakeshoe was referred to in the paragraph immediately preceding the heading Brake-Lining Wear Proportional to Work Done and also in the fourth paragraph under Actuation of Point at Toe of Shoe. Actual results indicate, however, that the actuation-factor as derived by these formulas is practically identical with the actual actuation-factor, and that the shoes will lock when the coefficient of friction is equal to the reciprocal of the actuation-constant Q. This would prove that the stress-strain relation of a drum, lining and shoe, when used in combination, must be very nearly elastic in characteristic, that is, the pressure must be very nearly proportional to the deflection or radial motion. As in most engineering, theoretical analysis is most useful as a general guide and as a basis for comparison and is not a substitute for the usual procedure of experimentation and guidance by the result of actual tests.

Replying further to Mr. Burkhardt's discussion, I can say that the brakes referred to did not give any trouble due to squeaking or chattering. Though truck and motor-coach brakes are far more rigid than those of passenger cars, I doubt whether a very great difference exists when their rigidity with respect to the weight of the vehicle handled and the consequent pressures and stresses are considered. Undoubtedly, as stated in Mr. Burkhardt's discussion, when chattering does occur, it can be traced to lack of rigidity and may be cured by adhering to the limitations of the arc of contact prescribed by his graphical method.

It might also be well to point out the fact that, when the actuating means are floating, the self-actuating shoe will have a greater total normal-pressure than the non-self-actuating shoe. The reaction balancing this difference is taken by the flange of the brake drum, which is bolted to the wheel or hub, resulting in less distortion or out-of-roundness of the drum than were the entire reaction taken by the opposing shoe. If the shoes are not opposed, as is the case in the brakes illustrated in Figs. 5 and 6, practically the entire reaction to the normal pressure on both shoes will be taken by the flange bolted to the hub or wheel, and the preceding statement is therefore true of such a brake to an even greater degree, consequently the out-of-roundness of a brake drum of a given rigidity for a given braking-action is far less in the type of brake illustrated in Figs. 5 and 6 than in the type in which the shoes are diametrically opposed.

I agree with Mr. Burkhardt's statement that contact will take place at the ends of the shoes, as this should be the case even without distortion; but I cannot agree that the normal pressures at the ends can exceed the normal pressure at other points of greater radial motion, unless the brake drum is almost as flexible as a band-saw blade, in which event it could not be used because of other limitations, such as the loss of pedal travel. If the brake is of a very light construction, the shoe itself undoubtedly will share in the general lack-of-rigidity characteristic. Accordingly, a large part of the danger of higher pressures at the toe would be counteracted by the greater relative deflection of the shoe at the toe for a given unit-pressure, as compared to the deflection at other points on the shoe. The heel of the shoe is, of course, far more rigid, but, for a point on the heel to become self-locking as a point, the shoe would have to extend much farther than is the case in most designs. Referring to the brakeshoe illustrated in Fig. 2, and assuming a coefficient of 0.4, such a point at the heel of the shoe, to become self-locking as a point, would be located at an angle $\beta = 165$ deg. 20 min. I use the expression, "self-locking as a point," to emphasize the fact that the self-actuating tendency imparted to a shoe by a point on it, is not measurable by the self-actuation of that point considered as a point alone, but must first be modified by the ratio of its unit normal-pressure to the unit normal-pressure of all other points on the shoe. In other words, the effect of the actuation at any point is simply the tendency of the frictional force at that point to rotate the brakeshoe about its anchor-pin center, and, in the case of the self-actuating shoe, adds to the torque about the anchor-pin exerted on the brakeshoe by the cam. This increase in torque is resisted by increased normal-pressures at all other points as well.

If a shoe is doing any braking, it is because there is pressure between the lining and the drum brought about by a torque that tends to swing the shoe about its anchor-pin into the drum. The sources of this torque are the cam pressure and the frictional forces about the anchor-pin. If the actuation-factor is, let us say, 0.75, its significance is that, in the case of the self-actuating shoe, 0.75 of the total torque about the anchor-pin is supplied by the frictional forces and 0.25 is supplied by the cam; in other words, the total actuating-torque on the shoe is four times as great as the cam torque alone. Assuming the same actuation-factor, in the case of the non-self-actuating shoe, the significance is that the total torque on the shoe is less than the cam torque alone and that the total torque is $1/(1+0.75)$ or $4/7$ of the cam torque. The total normal-pressure, and, consequently, the total frictional-force or braking-effort of a shoe, is proportional to the algebraic sum of the cam and frictional torques acting about the anchor-pin center.

I am very much indebted to Mr. Burkhardt for his very constructive criticism, and I hope and believe that the resulting discussion has cleared up some important points which, unfortunately, were not given proper consideration in my paper.

Personnel of 1926 Standards Committee

THE accompany.ng personnel of Divisions of the Standards Committee practically completes the personnel for 1926. The Standards Committee this year consists of about 275 members grouped into 25 Divisions classified according to type of vehicle and major automotive component parts, units and materials. Subdivisions are very frequently organized under the Divisions to handle specific projects, particularly the more important ones, and in some cases to cooperate with similar committees of other organizations.

This year the Society is entering into the field of production work by establishing the Production Division to accomplish for the production interests in the automotive industry what the Standards Committee has done for the engineering groups for many years. The Division will be comprised of men who are primarily production experts and the work will relate to production equipment and methods.

A consolidation of groups in the Standards Committee work has been accomplished by discontinuing the Frames, Nomenclature and Springs Divisions whose work was limited to closely defined channels and it was believed could be better handled by the vehicle and other more generally representative Divisions. The Radiator and Stationary Engine Divisions were also discontinued and their work will be consolidated under the Engine Division. The Engine Division as now organized will be representative of the various types of internal-combustion engines, excepting motorcycle, and the engine-cooling group, provision being made for the organization of Subdivisions of the Engine Division to handle subjects relating principally to particular types of engine. The groups represented include the aeronautic, automobile, motorcoach and motor truck, tractor, stationary including those for isolated electric-lighting plants, motorboat and Diesel types.

Special and cooperating committees are appointed either temporarily or permanently to handle specific subjects or such subjects as may not come within the classifications of Divisions. Their reports as a rule are eventually passed on by the Standards Committee.

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EXPORTS TO LATIN AMERICA

A STATISTICAL review of our present export position in Latin America is decidedly reassuring. In 1913 the United States supplied 24 per cent of the total imports of the 18 more important Latin American countries. In 1923 our share of these imports had advanced to 40 per cent; and such figures as are available for 1925 indicate even further improvement in this gratifying gain. Preliminary estimates of the total exports of the United States to Latin America for the calendar year of 1925 indicate aggregate shipments valued at $865,000,000. This total represents an increase of about 13 per cent in the value of exports to this region over the year 1924, and is about two and one-half times the average pre-war exports to that region. This advance in our export shipments to Latin America was common to all sections of that area, but it is particularly significant that the more highly competitive markets of South America led in this increased business, with the less difficult trade areas of Central America, Mexico and the West Indies following in the order mentioned.

In 1924 Argentina purchased automobiles valued at about $16,000,000, most of which came from the United States. This was more than three times the value of her automobile imports in 1913. In 1913 we sold in Latin America automobiles valued at $4,652,000; in 1924 our total automobile sales to Latin America aggregated $55,654,000. Our increase in sales of commodities indicates that our progress in Latin America is being made more by the creation of new markets than by the diversion of previously existing markets to us at the expense of Europe.

Ten years ago our total foreign investments were estimated at about $2,000,000,000; but at the end of 1924 the total American investments abroad, excluding advances made by our Government, had increased to $9,090,000,000, of which $4,040,000,000, or 44 per cent, has gone into Latin America, and 80 per cent of these investments in Latin America are in productive enterprises, the upbuilding of her economic power, the development of her vast resources. In the first 10½ months of 1925, $216,591,000 of America capital went into the further expansion of Latin American industry and economic wellbeing.

Some of our exporters, entering agency relations in Latin America without adequate information, have given exclusive representation to firms without facilities to cover a large area or to back up their American lines with proper service. There is now little excuse for lack of knowledge in these particulars. The Department of Commerce, in collaboration with the Consular Service, maintains an organization in Latin America and in the City of Washington equipped to provide exactly this kind of information to interested American commercial organizations.

All gains in the world's markets must be made on the basis of continuing service, if they are to be held permanently. This is discouraging speculative enterprise and encouraging the entrance into this great Latin American market of the best elements in our commerce and industry. And they are proceeding on the basis that trade with our Latin neighbors must advance on the basis of reciprocal advantages; that with every upward lift of American sales or investment there must be commensurate improvement in the economic wellbeing of the people of that great area.—From an address before the New England Export Club, by Dr. Julius Klein, Department of Commerce.

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Sleeve-Valve Engines

By P. M. HELDT[1]

DAYTON SECTION PAPER *Illustrated with* DRAWINGS AND PHOTOGRAPHS

ABSTRACT

FIRST sketching the history of the sleeve-valve engine, the author reviews the valve action of the Knight, discusses combustion-chamber shape and comments upon permissible compression, remarking also on the subject of carbon deposit in the sleeve-valve type of engine. Endurance tests of Knight engines are described and, in the author's opinion, should constitute a reliable guide in judging its performance characteristics. From the beginning, one of the foremost claims for the sleeve-valve engine has been that the sleeve type of valve permits much greater port openings and more rapid opening and closing of the ports. In view of this claim, it is said to be rather strange that sleeve-valve engines have not been more of a factor in speed contests; but the explanation undoubtedly is that exceedingly large valve-capacity can be obtained with poppet valves if quiet valve-action is not a consideration, according to the author.

Considering the timing of sleeve valves, it is stated that Knight sleeve-valve engines generally have a more extreme timing than poppet-valve engines; that is, the lead of the exhaust-valve opening and the lag of the inlet-valve closing are greater. The foregoing statement is enlarged upon. A recent development of Knight-engine practice in Europe has been the substitution of light steel-sleeves for the original cast-iron sleeves, which has made possible much higher engine-speeds. Illustrations of this practice are presented and the effect upon engine performance is discussed. Eventually, the wide junk-ring on the cylinder-head will be dispensed with, according to the author, and his reasons are given. Other subjects treated in detail are the balance of sleeves, the amount of power required to operate the sleeves, detonation characteristics and lubrication requirements of the sleeve-valve type of engine, the desirable clearance between sleeves and the circulation through the head-jackets. An outline is then given of the types of Knight engine developed by the Mercedes company, with illustrations.

Discussing single-sleeve engines, it is said that a single sleeve cannot perform the functions of both an inlet and an exhaust valve if endowed with merely a simple reciprocating motion, and the Burt sleeve therefore has a combined reciprocating and oscillating motion imparted to it, and this is explained in connection with a diagram. Regarding the form of ports desirable, the general form is not unlike that of a right-angle triangle, with the base up in the case of the cylinder-wall ports and down in the case of the sleeve ports. Studies of the valve action and of the port action are made, with accompanying diagrams, being inclusive of both inlet and exhaust ports. The double port in the sleeve is also described and illustrated, and the Vauxhall single-sleeve engine is explained in detail, three views being shown.

In conclusion, the author says that just as competition is the life of trade, so is competition a stimulant to technical progress. It is very doubtful whether poppet-valve engines would be so silent in operation as they are today if it had not been for the advent of the Knight engine; at any rate, progress in silencing that type of engine would not have been nearly so rapid. The introduction of the single-sleeve-valve engine on the American market may introduce another element of competition which will stimulate further development in the older types.

OF the many types of sleeve-valve engine patented and experimented with, only two have come into extensive commercial use, the double-sleeve type invented by Charles Y. Knight and the single-sleeve type invented by Peter Burt. Knight was the pioneer of the sleeve valve. He began his work along this line in Chicago in 1905 and built a car equipped with his engine. At that early period in automobile history the great majority of poppet-valve engines were very noisy in operation, even though their operating speeds were moderate, compared with present-day standards; the points in the design of cam mechanism which accomplish silent operation were not well known as yet, in addition to which the central portion of the valve train, including the gap between the valve-stem and tappet, were always exposed. In fact, in the British Daimler poppet-valve engine, which was the first to be replaced by the sleeve-valve type, the cams themselves were located outside the crankcase; so the benefit of an oil cushion on the surfaces in contact was not obtained. Knight's primary object was to render the engine more quiet in operation, by making the valve action positive, and he emphasized this by referring to his engine as the Silent Knight.

Attempts to interest American automobile builders in the engine having failed, Mr. Knight went abroad and succeeded in selling his British rights to the Daimler Motor Co., Coventry, England, which occupied an important position in the British motor-vehicle industry. At about the same time the Daimler Motor Co. of Cannstadt, often referred to as the Mercedes Co., secured the German rights; the Minerva Co., of Antwerp, the Belgian rights and Panhard & Levassor, of Paris, the French rights. These various firms all were preeminent in the automobile industries of their respective countries, and their decisions to adopt the Knight engine naturally created a deep impression throughout the automobile world. The British firm soon discontinued production of the poppet-valve engine and exhibited at the London Show in the fall of 1908 three models equipped with Knight engines, while the German Daimler and the Panhard firms adopted the sleeve-valve engine for part of their production only.

After his marked success in Europe, Knight returned to this Country and sought to interest builders of automobiles in his patents, the plan being to license four companies, instead of a single one, as in each of the European countries. The four licenses were eventually placed with the F. B. Stearns Co., the Columbia Motor Car Co., the Stoddard-Dayton Motor Car Co., and the Moline Automobile Co. More recently the four licenses have been controlled by the Willys-Overland Co., the F. B. Stearns Co., the Sterling-Knight Co., and the Yellow Sleeve Valve Engine Works, Inc. The largest producer of Knight engines in this country is the Willys-Overland Co. Until recently, this company turned out a four-

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cylinder car having a sleeve-valve engine, and at present is producing two models of Knight-engined sixes. The four-cylinder Knight-engines of the Willys-Overland Co. are being used also in commercial vehicles, on light trucks built by the Federal Motor Truck Co.

The Yellow Sleeve Valve Engine Works, Inc., which is successor to one of the original Knight license holders, the Moline Automobile Co., has been building Knight sleeve-valve engines for the Fifth Avenue Coach Co. of New York since 1915 and, at present, is producing motor-coach engines for the associated Yellow Coach Mfg. Co.

Knight's striking success caused many other inventors to turn their attention to positively operated types of engine valve and, during several years from 1910 on, there was a flood of sleeve, rotary and piston-valve engines or engines with positively operated valves. A few of these engines were backed by ample capital and engineering talent and some of them, like the Itala rotary-valve engine and the Mead, used in the Speedwell car, were built for a number of years; but, so far as I know, the only other non-poppet-valve engine that has endured is the single-sleeve-valve engine of Burt & McCullum, which has been manufactured since 1911 by a Scottish firm, the Argyll Motor Car Co., of Alexandria near Glasgow. Recently the Vauxhall Co., another old established firm in the British motor-car industry and which is now controlled by the General Motors Corporation, has taken up the Burt engine, which has also been built on a small scale by Picard & Pictet of Geneva, Switzerland, for use in a car known as the Pic-Pic, and by the French racing driver Guyot. The American rights to this engine have been acquired by the Continental Motors Corporation; therefore, one does not have to be much of a prophet to foresee for it an important role in American automotive developments the next few years.

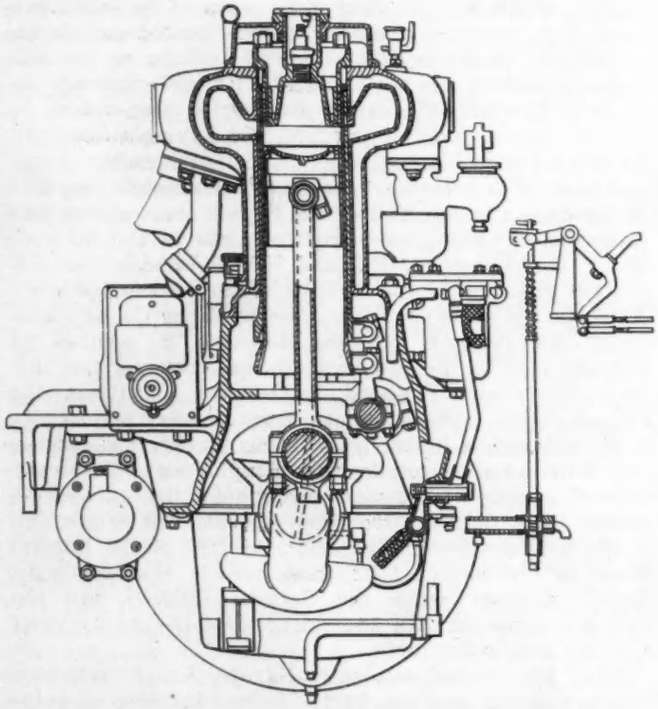

FIG. 1—PRINCIPAL FEATURES OF THE KNIGHT ENGINE

The Valve Functions Are Performed by Two Concentric, Ported Sleeves, Generally of Cast Iron, Which Are Inserted Between the Cylinder-Wall and the Piston. The Sleeves Are Given a Reciprocating Motion by Connection to an Eccentric Shaft Driven from the Crankshaft through the Usual 2 to 1 Gear, Their Stroke, in the Older Designs at Least, Being Either 1 or 1⅛ In. The Sleeves Project from the Cylinder at the Bottom and, at the Top, They Extend into an Annular Space between the Cylinder-Wall and the Special Form of Cylinder-Head So That, during the Compression and the Power Strokes, the Gases Do Not Come into Contact with the Cylinder-Wall But Are Separated Therefrom by Two Layers of Cast Iron and Two Films of Lubricating Oil. The Cylinder, As Well As Each Sleeve, Is Provided with an Exhaust-Port on One Side and with an Inlet-Port on the Opposite Side. The Passage for Either the Inlet or the Exhaust Is Open When All Three of the Ports on the Particular Side Are in Register with Each Other

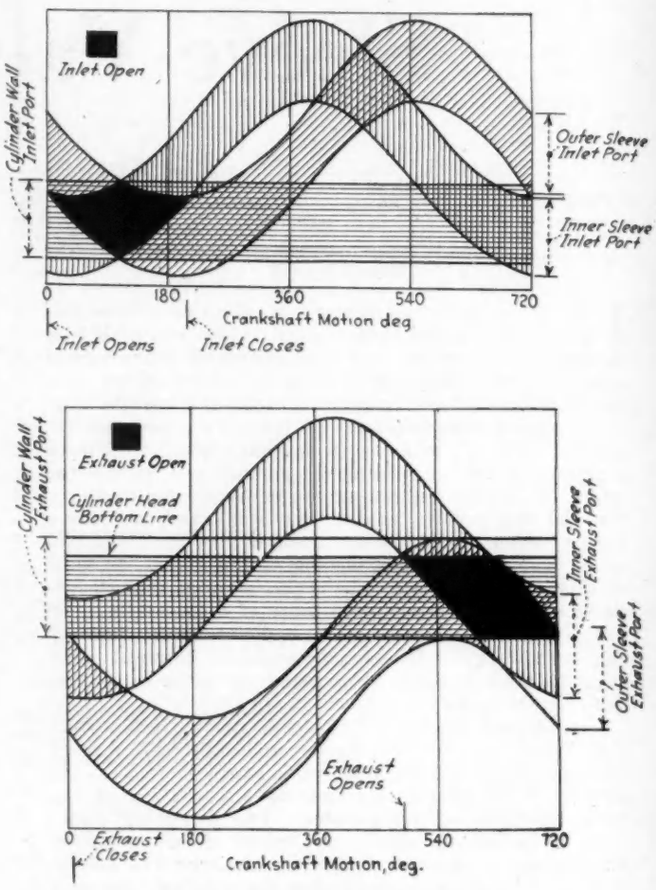

FIG. 2—SLEEVE-MOTION DIAGRAMS

For Studying the Valve or the Port Action of a Sleeve-Valve Engine, the Curves Can Be Laid Out with the Aid of the Formula for Crank-Train Motion. To Secure the Needed Degree of Accuracy, It Is Well To Lay the Curves Out on a Large Scale and Account Must Be Taken of the Fact That, Whereas the Strokes of Both Sleeves Are Alike, the Lengths of Their Connecting-Rods Generally Are Different

As regards the patent situation, it may be cited that, shortly after the single-sleeve-valve engine was placed on the market in Great Britain, there was a patent contest between Knight and Argyll; but it was held by the court that the Burt engine was not an infringement of the Knight patents. The first of the Knight United States patents will expire in August, 1927, but a number of other patents exist and I see no immediate prospect that the double-sleeve-valve field will be thrown wide open at that time.

Among other sleeve-valve engines on which patents have been taken out, one with a rotary sleeve, patented by Renault in France but which was never placed in production, may be mentioned. A similar engine was under development in this Country during a period of several years, but no serious effort was made to build it. The single sleeve between cylinder-wall and piston was rotated by helical gearing at one-quarter crankshaft-speed.

VALVE ACTION

Although the Knight engine is generally well known, I will review its principal features briefly. Referring to Fig. 1, the valve functions are performed by two con-

centric, ported sleeves, generally of cast iron, which are inserted between the cylinder-wall and the piston. The sleeves are given a reciprocating motion by connection to an eccentric shaft driven from the crankshaft through the usual 2 to 1 gear, their stroke, in the older designs at least, being either 1 in. or 1 1/8 in. The sleeves project from the cylinder at the bottom and, at the top, they extend into an annular space between the cylinder-wall and the special form of cylinder-head so that, during the compression and the power strokes, the gases do not come into contact with the cylinder-wall but are separated therefrom by two layers of cast iron and two films of lubricating oil. The cylinder, as well as each sleeve, is provided with an exhaust port on one side and with an inlet port on the opposite side. The passage for either the inlet or the exhaust is open when all three of the ports on the particular side are in register with each other.

The valve or port action of such an engine can be studied to advantage by a sleeve-motion diagram, such as those shown in Fig. 2. The curves can be laid out with the aid of the formula for crank-train motion; but, to secure the necessary degree of accuracy, it is well to lay them out on an enlarged scale and account must be taken of the fact that, whereas the strokes of both sleeves are alike, the lengths of their connecting-rods generally are different. The inlet begins to open when the bottom edge of the port in the outer sleeve, moving downwardly, passes the top edge of the port in the inner sleeve, also moving downwardly but at a slower rate. The inlet port closes when the bottom edge of the port in the inner sleeve, moving upwardly, passes the top edge of the port in the outer sleeve, also moving upwardly. This passing of the edges of the ports in the sleeves takes place while they are opposite the inlet port in the cylinder-wall. The inlet ports of the two sleeves also come into registry about 360 deg. of crank motion later, when the piston has started on its power stroke, but these ports are then high up between the cylinder-head and the cylinder-wall, and the port in the inner sleeve is sealed by the junk-ring in the cylinder-head.

Combustion-Chamber Shape

One advantage of the sleeve-valve engine that is immediately apparent upon an inspection of the sectional view is the favorable form of the combustion-chamber. Of the heat energy released by the combustion, a large proportion ordinarily passes into the cooling-jacket, and this proportion naturally varies with the ratio between the cooling area and the volume of the combustion-chamber. It is particularly important to have the cooling area as small as possible during the early part of the power stroke, as it is then that the working fluid is at its highest temperature and loses energy to the jacket most rapidly.

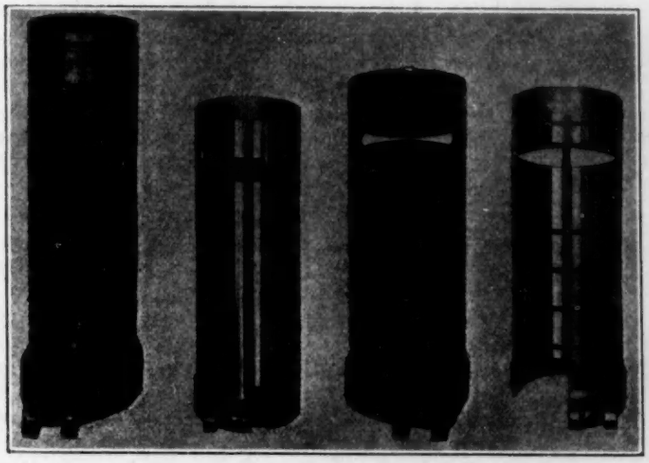

Fig. 4—Light Steel-Sleeve for Knight Engine
Recent European Practice Has Been To Substitute Light Steel-Sleeves for the Original Cast-Iron Sleeves, Which Has Made Possible Much Higher Engine-Speeds

The most favorable form of combustion-chamber from this point of view is a sphere and, in the Knight engine, with its domed cylinder-head and the dished piston-head used in some designs, the spherical form is approached more closely than in any other type of engine, even more closely than in engines with valves set at an angle in the head, as these usually have plane piston-heads. Determinations of the percentage of the heat energy passing into the cooling water in Knight engines have confirmed this conclusion based on theoretical considerations. Measurements made by Professor Riedler, of the Charlottenburg Technical College, on three different models of sleeve-valve engine, showed the loss to the water-jacket to be from 19.5 to 25.8 per cent; whereas, the lowest loss to the jacket measured on poppet-valve engines by the same investigator was 30.7 per cent. Of course, it must not be assumed that all the reduction in the loss to the water-jacket is added to the useful output of the engine. If the cooling area is small, the heat is retained in the cylinder; this helps to maintain the pressure throughout the stroke and thus adds to the output of mechanical work, but there is also more heat energy left in the gases at the end of the power stroke, which is discharged with the exhaust.

Conservation of heat in the working fluid during the

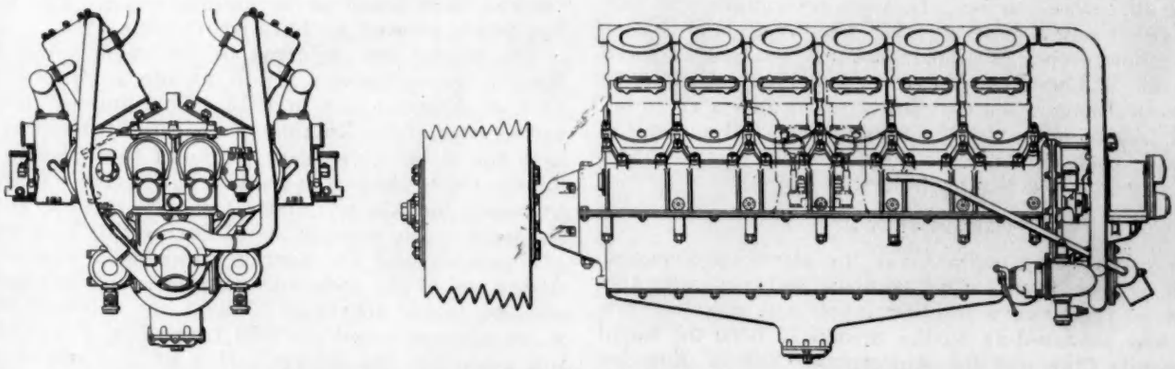

Fig. 3—The Panhard 12-Cylinder Sleeve-Valve Aircraft Engine
In France, the Panhard Company Developed and Patented an Engine Design Embodying Light Steel-Sleeves, Which Are Used in Both Its Aircraft and Its Passenger-Car Engines

power stroke is due not only to the relatively small extent of the combustion-chamber wall-surface, but also to the special character of this wall surface, which makes heat-flow through it more difficult. At the beginning of the power stroke, which is the most important point from the standpoint of heat-loss to the jacket, less than one-half of the combustion-chamber wall-surface is directly water-cooled, this being the under side of the cylinder-head. The piston-head, which forms a greater percentage of the total wall-surface than in most engines because of its dished form, is far removed from the jacket-water, measured in terms of resistance to heat-flow. Comparatively little of the inner sleeve is exposed to the working fluid at this time, but whatever heat this surface absorbs must pass through three thicknesses of metal and two films of oil before it reaches the water. Just how much this insulating effect, particularly that of the oil-films, retards the heat-flow can only be conjectured, but it is undoubtedly a factor in the reduction of the full-load jacket-losses in the Riedler test, from an average of about 34 per cent in poppet-valve engines to an average of 23.5 per cent in sleeve-valve engines.

Permissible Compression

The slower rate of heat-flow to the jacket might be expected to have undesirable results in causing overheating, with its associated troubles of faulty lubrication, preignition and detonation. In the conventional engine, three points usually reach temperatures above the average of the combustion-chamber wall; namely, the exhaust-valve head, the spark-plug and the piston-head. The sleeve-valve engine has no exhaust-valve head to cause trouble. The spark-plug is usually well cooled, being screwed into a boss located centrally in the cylinder-head and completely surrounded by water. Thus there remains only the piston-head, and no particular difficulty seems to be experienced in keeping this below the temperature at which danger of self-ignition begins. The suggestion has been made that, owing to the restrictions in the paths of heat-flow, it might be necessary to operate sleeve-valve engines at lower compression, but the compression-ratios actually used are substantially the same as those used in poppet-valve engines. Thus, various Stearns' models have compression-ratios of 4.5, 4.6 and 4.9 and the Sterling-Knight compression ratio is 4.5. In the Vauxhall single-sleeve-valve engine the volumetric compression-ratio is 5, which is probably higher than the ratio used in any other automobile engine.

In the earliest Knight engines there was evidently some trouble from inadequate cooling, or at least the designers felt they had to provide against such trouble. Recourse was had to an expedient that is also made use of in air-cooled engines, that of restricting the inlet passage at some point so that the maximum rate of charge flow occurs at a comparatively low engine-speed. Thus, in a Coventry Daimler engine investigated by Professor Riedler, the carbureter choke had a cross-sectional area of only 2.05 sq. in. while the full opening of the inlet port was substantially 4.00 sq. in. This for an engine with 4 x 5-in. cylinders.

Carbon Deposits

It also has been claimed for the sleeve-valve engine that it is free from carbon troubles, but just why this should be true has never been explained clearly. The claim was bolstered-up by the reports of both the Royal Automobile Club and the Automobile Club of America endurance tests, to which reference will be made later, in which statements occurred to the effect that examination of the working parts after the tests had shown practically no carbon accumulations. Since these tests were made, much additional experience has been accumulated, however, and it is now fairly evident that what was then taken to be a characteristic of the sleeve-valve engine was probably more a characteristic of the particular oiling-system used. That sleeve-valve engines will form carbon deposits in the combustion-chamber is evidenced by the fact that one builder of such engines issues very specific instructions regarding carbon removal. The carbon must under no circumstances be burned-out, as this may injure the sleeves, and it must not be scraped from the sleeves, because carbon on the sleeves is said to be an advantage. One other consideration which is not mentioned but which may have had an influence in making the recommendation is that scraping off carbon would not improve the seal of the sleeves nor their bearing characteristics. Of course, this practically amounts to saying the same thing in other words.

Endurance Tests of Knight Engines

Aside from the tests made on sleeve-valve engines by Professor Riedler and reported by him in his book on the Scientific Determination of the Merits of Automobiles, at least three extended, searching public-tests of the Knight engine have been made, the results of which should constitute a reliable guide in judging its performance characteristics. The first of these was conducted in 1909 by the Royal Automobile Club of Great Britain, on a four-cylinder Daimler engine of 96-mm. bore and 130-mm. stroke, approximately 3¾ x 5 in. The engine was run on the bench continuously for 5 days, 12 hr. and 58 min. at a piston speed of 1000 ft. per min., which was about normal for that period, developing an average of 38.83 hp. According to the rules of the test the output was to be more than 1.3 times the horsepower rating under the Royal Automobile Club formula, this rating being 22.85 hp. The fuel-consumption was at the rate of 0.668 lb. per hp-hr. Two stops of a total duration of 17 min. were made, but they were such that no penalty was incurred. At the completion of this test the engine was put into a car and driven a total of 2143 miles, mainly on Brooklands track, at an average speed of 41.88 m.p.h. The weight of the car with load was a little more than 3300 lb. during some of the runs, and 3600 lb. in others. This was followed by a final bench-test of 5-hr. duration, in which the engine developed an average of 38.96 hp., with a gasoline consumption average of 0.677 lb. per hp-hr. In their report on the tests, the judges remarked at the conclusion of the final bench-test the engine was dismantled and no wear on any of the fitted surfaces was perceptible. The cylinders and pistons were found to be notably clean. The ports of the valves showed no burning or wear.

The second test referred to was made on a Moline-Knight four-cylinder 4 x 6-in. engine at the Automobile Club of America late in 1913. This engine, unlike the earlier ones of the Knight type, had full force-feed oiling, even up to the piston-pins, through the holes in which oil was fed to the piston guide. The engine was run continuously for 336 hr., under full-open throttle and with set spark. The average speed during this long run was 1117 r.p.m., and the average brake-load was 36.5 hp. At the end of the endurance run the speed was increased and the engine developed 53 b.hp. for a period of 1 hr. at an average speed of 1678 r.p.m. In a special 5-hr. run made for the determination of fuel efficiency, the fuel consumption was at the rate of 0.63 lb. per hp-hr., with an engine output of 39.8 hp. at 1114 r.p.m. The

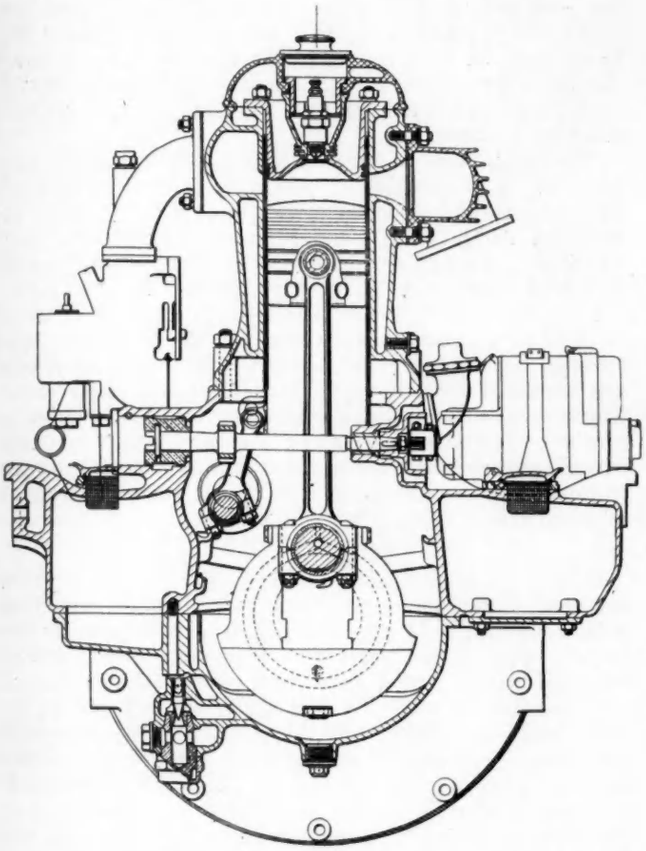

FIG. 5—SECTIONAL VIEW OF WILLYS-KNIGHT ENGINE
The Drawing Shows Pistons, Sleeves, Cylinder-Heads, the Unusual Type of Junk-Ring, Separate Cylinder and Water-Jacket Heads and the Spark-Plug Located in a Well

volumetric efficiency when running under full throttle was determined to vary from 78 per cent at 900 r.p.m. to about 70 per cent at 1600 r.p.m.

Regarding the condition of vital parts of the engine at the end of the test, the report stated that these were,

> . . . without exception, in excellent condition. There was no perceptible wear on the bearings, sleeves or other parts. The slight irregularities in the sleeves were built-up with carbon to form close-fitting glossy-surfaces. The ports in the sleeves were not burned, and there was only a very slight deposit of carbon on the port edges. The cylinder-heads and the tops of the pistons showed only a very thin coating of carbon, and only small quantities of carbon were found elsewhere. No play could be found in any bearing, and there was every indication of perfect lubrication . . .

The most recent test of a Knight engine under severe conditions was of a Panhard-Knight aircraft-engine by the French Air Service in a competition in which prizes to the amount of 2,000,000 francs were offered by the Under-Secretary for Air. A 12-cylinder Panhard-Knight was one of 13 engines entered and one of 4 that completed the test successfully. This is one of the first sleeve-valve engines ever built for aircraft purposes (See Fig. 3). In this test a limit of 7.25 lb. was placed on the weight per brake horsepower, inclusive of fuel, oil and water, sufficient for a 5-hr. flight. A few data concerning this engine may be of interest. The 12 cylinders each have a bore of 140 mm., (5.51 in.), and a stroke of 170 mm., (6.69 in.), and are arranged V-fashion with an angle of 60 deg. between the two sets. The volumetric compression-ratio is 5.4, which is about the same as that used with other aircraft engines designed to operate on aviation gasoline. One eccentric shaft is provided for each of the two sets of cylinders. Starting is by compressed air admitted to the cylinders through a distributor mounted on the engine. The rated output is 550 hp. at 1800 r.p.m. and the weight in running order, 1155 lbs. or 2.1 lb. per hp.

PORT-OPENING AREA

From the beginning, one of the foremost claims for the sleeve-valve engine has been that the sleeve type of valve permits much greater port openings and more rapid opening and closing of the ports. In view of this claim, it is rather strange that sleeve-valve engines have not been more of a factor in speed contests. In fact, up to the time of the recent Panhard and Voisin successes on the Monthlery track in France, all speed records stood to the credit of the poppet-valve engine. The explanation undoubtedly is that exceedingly large valve-capacity can be obtained with poppet valves if quiet valve-action is not a consideration. An investigation made in connection with the development of the Liberty-12 aircraft-engine showed that, whereas a poppet valve theoretically attains its maximum capacity when the lift is made equal to one-quarter the clear diameter, actually, the capacity corresponding to such a lift is only about two-thirds the maximum and the flow through the valve will increase with the lift far beyond this point. In passenger-car engines, owing to the requirement of silent operation, the lift is generally made materially less than one-quarter the clear diameter and, moreover, the valve is lifted very gradually. In racing engines, on the other hand, the height and the rate of lift are made as great as the physical qualities of the material involved will permit, and this allows a large increase in output.

At moderate speed, the specific output of sleeve-valve automobile-type engines is apparently about 15 per cent greater than that of poppet-valve engines. The torque curve of a Yellow Sleeve Valve six-cylinder 4¼ x 5½-in. motorcoach-engine, which has been furnished to me by the builder, shows a brake mean-effective pressure varying from 107 lb. per sq. in. at 800 r.p.m., to 80 lb. per

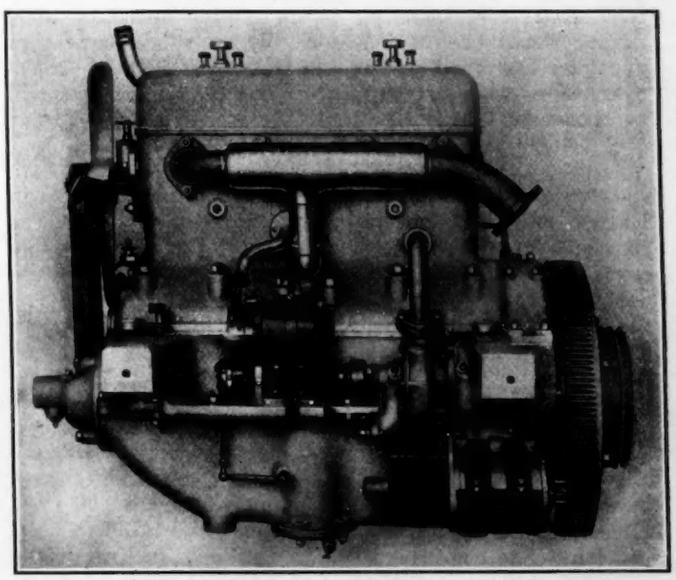

FIG. 6—LATEST MERCEDES-KNIGHT ENGINE
This Is a Four-Cylinder-Block Engine and Has the Upper Portion, But Not the Upper Half, of the Crankcase Cast Integrally with the Cylinders

sq. in. at 2250 r.p.m., the speed of maximum output. This compares with a brake mean-effective-pressure range for the average passenger-car engine of from 85 to 70 lb. per sq. in. On the other hand, aircraft engines, and presumably racing automobile engines also, develop a higher brake mean-effective pressure than 107 lb. per sq. in.

Timing of Sleeve Valves

It is worthy of note in this connection that Knight sleeve-valve engines generally have a more extreme timing than poppet-valve engines; that is, the lead of the exhaust-valve opening and the lag of the inlet-valve closing are greater. In the Yellow Sleeve Valve Engine Works motorcoach engine, for instance, the exhaust opens 52 deg. ahead of bottom dead-center and the inlet closes 60 deg. past bottom dead-center, while the exhaust closes 11 deg. past top dead-center and the inlet closes 7 deg. past top dead-center. In addition to the large port-openings obtainable, the short and direct path of the incoming and outgoing gases should be a factor in enabling the sleeve-valve engine to maintain its volumetric efficiency at high speed. It is a well-known fact that any bends in a passage through which there is a rapid flow always introduce a considerable resistance to the flow, as, at such turns or bends, eddies are formed which cause a part of the energy of the flow to be dissipated.

One thing that distinguishes the sleeve-valve engine from the more conventional type is that its valves are positively operated and that the valve timing therefore is absolutely independent of the speed. However, with cast-iron sleeves, it was impossible to take full advantage of this feature, owing to the considerable weight of the sleeves and the consequent inertia forces. This is true more particularly when speeds such as 4000 r.p.m. are considered, which are now freely spoken of in connection with stock models in Europe. The cast-iron sleeves of the several engines on which test results are quoted in this paper weighed from 8 to 9 lb. each. Probably another pound can be added to allow for the inertia and centrifugal forces on the "eccentric" rod. Now the maximum inertia-force on a reciprocating weight of 10 lb. having a stroke of 1 in., at 2000 r.p.m. of the eccentric shaft, corresponding to an engine speed of 4000 r.p.m., would be about 700 lb., which would be rather great for the small bearings of the eccentric rods. Moreover, the sleeves are not in mechanical balance, as will be shown further on.

Steel Sleeves

A recent development in Knight-engine practice in Europe has been the substitution of light steel-sleeves for the original cast-iron sleeves, which has made possible much higher engine-speeds. As is well known, the tendency in Europe for many years has been toward engines of small piston-displacement and high piston-speed, and in this movement the Knight licensees could not compete successfully so long as they stuck to the cast-iron sleeve. The Panhard Co., in France, developed and patented a design embodying light steel-sleeves, which are used in both its aircraft engine and in its passenger-car engines. The sleeves are made of high-tensile-strength steel, and very thin, being 1.5 mm. (0.06 in.), as compared with 4 and 5 mm., (0.157 and 0.196 in.), in cast iron (See Fig. 4). In the older Panhard-Knight engines, as well as in others of the type, the combustion-chamber is sealed at the top by a single wide junk-ring and, according to the Panhard engineers, the very width of this ring made it difficult to effect a good seal. In the new design, four very narrow rings are used, and this made it possible to widen the ports greatly and thus to enlarge the port capacity. In the inner steel-sleeves, the ports are divided by numerous narrow bridges, so there is no possibility of the rings on the cylinder-head catching in them.

The limitations on the width of the ports having been removed, it is possible to maintain a high volumetric-efficiency up to very high speeds and in bench tests, the so-called 20-hp. four-cylinder engine of 105-mm. bore, (4.13 in.), and 140-mm. stroke, (5.51 in.), is said to have delivered nearly 200 hp. at 3400 r.p.m. At 1000 r.p.m., the output is said to be nearly 12 hp. per liter, or 61 cu. in., of displacement. Owing to the fact that steel on steel does not give a good bearing, the outer sleeves are provided with a very thin lining of white metal, which is applied to the sleeve by a centrifugal casting process. The Minerva Co., of Belgium, which has been experimenting with an eight-cylinder Knight aircraft engine, uses an outer sleeve of bronze together with an inner sleeve of steel.

Junk-Rings

It is my opinion that the wide junk-ring on the cylinder-head of Knight engines will eventually be dispensed with altogether. It was used by Knight because he employed ports in the valve sleeve of very great circumferential width, and a narrow ring would have dropped into these ports, as it were, or would have caught on the edges, with destructive effects. Bridges in cylinder-wall ports were commonly used at that time in two-stroke engines, and had been used also in auxiliary exhaust-ports at the bottom of the stroke in four-stroke engines, but Knight evidently feared that such bridges in the thin inner-sleeve without direct water-cooling would give trouble from overheating. Recent experience, however, has shown that there need be no such trouble.

Various modifications of the original form of junk-ring have been made. Thus, some builders have used a wide outer-ring of comparatively small radial thickness, which was pinned so that its gap could not come opposite the ports in the inner sleeve, and under it a number of narrow, stiffer rings, which furnished practically all the spring force. In the Willys-Knight engine, a wide junk-ring with internal flanges is being used (See Fig. 5). I understand that the chief reason for this construction is that it reduces wear on the sides of the ring-grooves in the cast-aluminum head, the lateral thrust on the ring being taken up by four or five instead of a single thrust-surface.

When the limitation placed by the junk-ring on the width of the ports is removed, the valve capacity can be increased greatly. In the older Knight engines, having a stroke of about 5 in., the width of the ports generally ranged between the limits of ½ and 5/8 in. If several bridges and narrow rings are used, this can be increased by at least 50 per cent and possibly doubled. Of course, the stroke of the sleeves has to be increased in the same proportion, and probably the real limit to the speed in such an engine is the inertia of the sleeves, which increases with the stroke.

Balance of Sleeves

A problem that deserves consideration when very high speed of operation is contemplated is that of the balance or unbalance of the sleeves. The dynamic forces on the pistons and the connecting-rods are the same in a sleeve-valve as in a poppet-valve engine and need not be considered here, but the sleeves introduce a new element of unbalance which cannot be entirely neglected. We know

that, in the conventional four-cylinder engine, the primary components of the unbalanced forces on the pistons and similar parts neutralize each other because the two end-pistons, Nos. 1 and 4, move up and down together and pistons Nos. 2 and 3 also move up and down together and always in opposition to pistons Nos. 1 and 4. Piston No. 4 follows piston No. 1 in its motion at a time interval corresponding to 360 deg. of crankshaft motion, and every phase of the cycle in cylinder No. 4 is behind the corresponding phase in cylinder No. 1 by this time interval. But 360 deg. of crank motion is the same as 180 deg. of eccentric-shaft motion; therefore, each sleeve in cylinder No. 4 follows the corresponding sleeve in

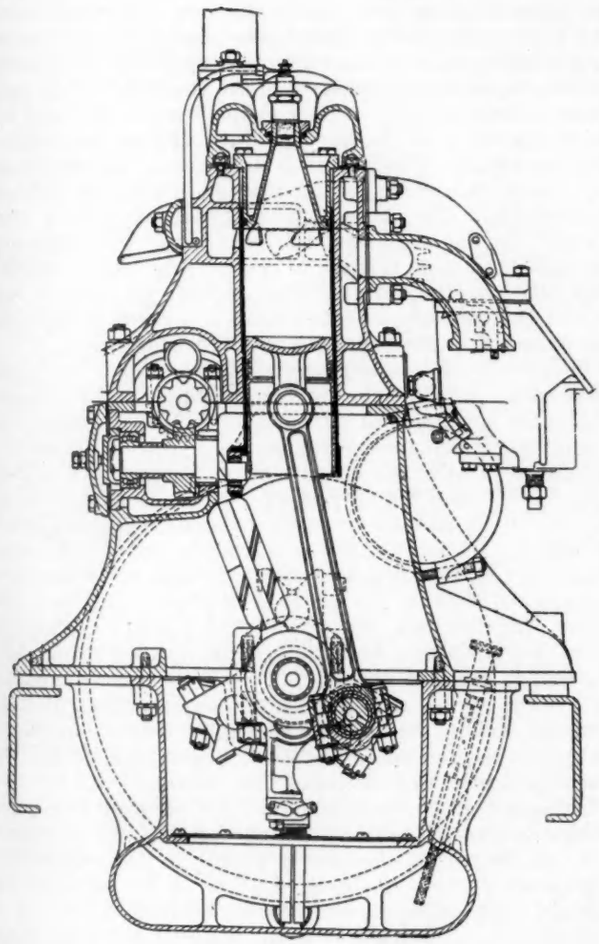

FIG. 7—CROSS-SECTION OF VAUXHALL BURT-TYPE ENGINE
Running Parallel to the Crankshaft in an Upper Compartment of the Crankcase Is a Shaft Which Is Driven from the Crankshaft by Gearing or by a Silent Chain, But at Crankshaft Speed Instead of One-Half That Speed as in the Conventional Engine

cylinder No. 1 at a time interval corresponding to 180 deg. of eccentric-shaft motion. From this it is evident that, when one of the sleeves in No. 1 cylinder is at the bottom of its stroke, the corresponding sleeve in No. 4 cylinder is at the top of its stroke. The primary unbalanced forces on these latter two sleeves are the maximum at this moment and, though they are opposite in direction, they cannot neutralize each other because of the fact that they act along parallel lines at a considerable distance from each other. The result is that a rocking couple in a vertical longitudinal-plane is introduced. At the same time the primary components of

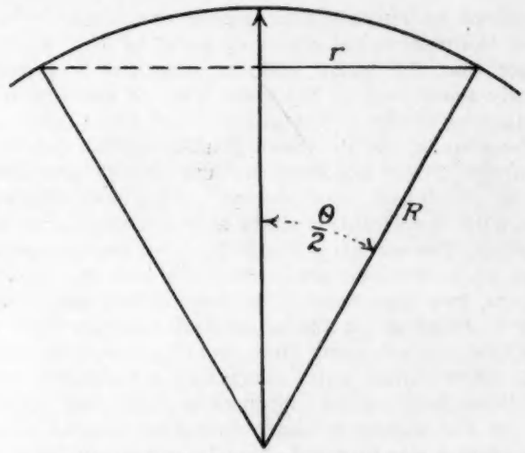

FIG. 8—DETERMINATION OF SLEEVE MOTION
This Diagram Is Used for Determining the Extent of the Oscillating Motion of the Sleeve

the unbalanced forces on the reciprocating parts of cylinders Nos. 2 and 3 are nil.

On the other hand, the secondary components of the unbalanced forces, which are the chief cause of vibration in a four-cylinder conventional-engine, are neutralized in the case of the sleeves. It was stated previously that the primary component of the unbalanced force on a sleeve in No. 4 cylinder lagged behind the same component of the same force in No. 1 cylinder by a time interval equal to 180 deg. of eccentric-shaft motion. Now the secondary component has twice the frequency of the primary component; that is, during the same time that the primary component passes through 180 deg., or half a cycle, the secondary component passes through a complete cycle. Hence the secondary component of the unbalanced force on a sleeve in No. 4 cylinder lags a whole cycle behind the same component of the same force in No. 1 cylinder, and therefore is in phase with it. The same relation therefore exists between the secondary components of the unbalanced forces on the sleeves in the different cylinders of a four-cylinder engine, as between the primary components of the unbalanced forces on the reciprocating parts of the piston train, and these secondary components neutralize each other.

POWER REQUIRED BY SLEEVES

One of the objections that have been made to the Knight engine in the past is that a great amount of power

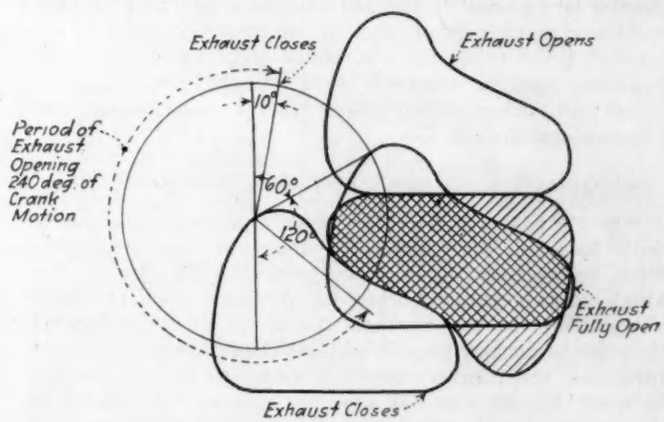

FIG. 9—THE BURT-TYPE EXHAUST-PORT ACTION
The Total of the Cross-Hatched Area Represents the Exhaust-Port in the Cylinder-Wall, While the Two Unsectional Areas Adjacent to It and the Similar Area Midway between Them and Registering with the Cross-Sectioned Area Represent the Exhaust-Port in the Sleeve in Three Different Positions

is required to operate the sleeves and that, in consequence, the mechanical efficiency must be low. Owing to the fact that the pump and the magneto or generator generally are driven by the same train of gearing, no determinations of the power absorbed by the sleeves alone have been made, but Professor Riedler made a determination of the power absorbed by the sleeves and the accessories combined. An electric motor was rigged up in line with the eccentric shaft and connected thereto by a coupling. The eccentric shaft thus had two independent sources of power and was driven by one or the other, or by the two together. The chain wheel had a slight degree of freedom on the shaft and was provided with an electric contact-point that would come into contact with another contact point carried on a rigid arm on the shaft if the chain ceased to drive the shaft, and the chain wheel on the eccentric shaft therefore lagged behind. This condition was brought about by gradually increasing the voltage applied to the terminals of the electric motor. When the contact was established, indicating that all the power for driving the eccentric shaft was being supplied by the electric motor, a lamp flashed and readings were taken of the electrical instruments in the motor circuit, from which readings the power required for driving the shaft could be calculated. A 96 x 130-mm. (3.79 x 5.12-in.) four-cylinder engine at 2000 r.p.m. developed a little more than 40 hp. on the brake and consumed exactly 2 hp. on the eccentric shaft. The power used to drive the shaft varied almost in direct proportion to the speed of the engine when operating under full throttle.

The power required for operating sliding sleeves is, of course, much greater than that required to operate poppet valves, but the total is so small that it does not lower the mechanical efficiency of the engine seriously. It has sometimes been argued that poppet valves must absorb considerable power because the exhaust-valves have to be lifted against a pressure of 50 or 60 lb. per sq. in. While it is true that this pressure exists in the cylinder at the beginning of the exhaust period when operating under full throttle, the valve does not have to be lifted against it any appreciable distance, for, immediately it is off its seat, the difference of pressure on its opposite sides is reduced greatly, and thus only a small amount of power is absorbed. The power required for compressing the springs is, of course, largely returned during the closing period of the valve. Probably of more importance than the loss in power due to sleeve friction in normal operation is the added starting friction. Owing to the large area of the surfaces in contact, it is easily conceivable that when the oil on these surfaces is congealed, their frictional resistance adds materially to the starting torque required, even though the motion is slow and the starting motor has a considerable mechanical advantage.

Detonation Characteristics and Lubrication

The claim has often been made that, with the sleeve-valve engine, there is less tendency to fuel-knock than with engines having side pockets. If the theory that knocking is dependent upon the distance from the point of ignition to the farthest point of the combustion-chamber is correct, then this claim should be well founded, for, with the compact form of the combustion-chamber in most Knight engines and the spark-plug located in the center of the cylinder-head, it is difficult to see how the most remote portion of the combustible charge could be brought much closer to the points of ignition. The absence of the equivalent of an exhaust-valve head should also be a help in this connection.

Lubrication

When the Knight engine was first placed on the market, many engineers professed themselves to be skeptical regarding the possibility of effectively lubricating the long sleeves under all conditions, but long experience has shown that their apprehensions were ill-founded and that all difficulties of lubrication can be overcome by careful design. The early Knight engines had splash lubrication, like practically all other automobile engines of the period. The lower ends of the working parts had oil thrown on them by dippers secured to the connecting-rods, and the vacuum existing in the combustion-chamber during the inlet stroke tended to draw the oil up toward the top of the sleeves. However, the suction in the combustion-chamber is greatest when the throttle is nearly closed and the engine is running under small or no load, when comparatively little lubricant is required; whereas, when the throttle is fully open, there is little vacuum in the combustion-chamber and less oil is drawn up to the upper ends of the sleeves, when a copious supply of lubricant is desirable. Knight sought to remedy this difficulty by making his splash troughs controllable. The troughs were provided with a pivot support on one side, and the other side was connected through a suitable linkage to the throttle lever in such a way that, as the throttle was opened, the trough was raised, the quill dipped into the oil to a greater depth and the splash, in consequence, was more energetic.

Practically all modern Knight engines have force-feed or pressure lubrication, similar to poppet-valve engines. The same fundamental difficulty of a lesser tendency of the oil to travel upward inside the cylinder under conditions of full-throttle operation exists with this system, and this is said to be one of the reasons that, until recently, little success was achieved with Knight engines in races. In racing, the throttle is kept wide open practically all the time, and the force tending to augment the upward flow of the oil is very slight. In fact, one driver who has had experience with Knight engines in speed work claims that, to get the most out of such engines, it is good practice to close the throttle partly for a short period at regular intervals; and that the loss of speed during these short periods is more than made up by the higher speed between, made possible by the resulting better lubrication of the sleeves.

To assure good distribution of the oil over the large surface of the sleeves and the surfaces of the cylinder-wall and the piston, the sleeves generally have numerous oil-grooves turned in them, and oil holes are drilled through, often along a helical line, so that oil can pass from one side of each sleeve to the opposite side. Between the inlet and the exhaust ports, grooves are often cut in an up-and-down direction to help prevent leakage between these ports. With pressure lubrication, provision generally is made to increase the supply of lubricant to the upper end of the sleeves under heavy-load conditions. A valve placed in the oil line is opened through connection with the throttle linkage as soon as the throttle is opened beyond a certain point. Oil is then delivered directly into the annular space between the cylinder-head and the cylinder-wall above the sleeves.

Clearance Between Sleeves

A point of detail that arises in the design of sleeve-valve engines is the clearance to be allowed between the various concentric parts. If the fit were made too tight, danger of seizing would exist, and the frictional loss would also be increased. On the other hand, if the fit

were too loose, piston-slap might be expected, especially as there are three clearances in series, all of which must be taken-up when the side thrust of the piston reverses in direction. Practices of the different builders in this respect differ to a certain extent. Some have used as small a clearance as 0.002 in. in an engine of about 4-in. bore. Others use a somewhat larger clearance, pointing out that the seal of the combustion-chamber is effected by flexible rings and is not dependent upon the closeness of the fit between the sleeves.

Throughout the power stroke, the inner sleeve is near the top of its stroke and both ports in it are far removed from the combustion-chamber and are sealed by the junk-ring. It is only during the earlier part of the compression stroke that there is comparatively little lap of the inner and outer sleeves at the inlet port, and this lap is partly opposite the port in the cylinder-wall. The effectiveness of the seal at this moment has some-

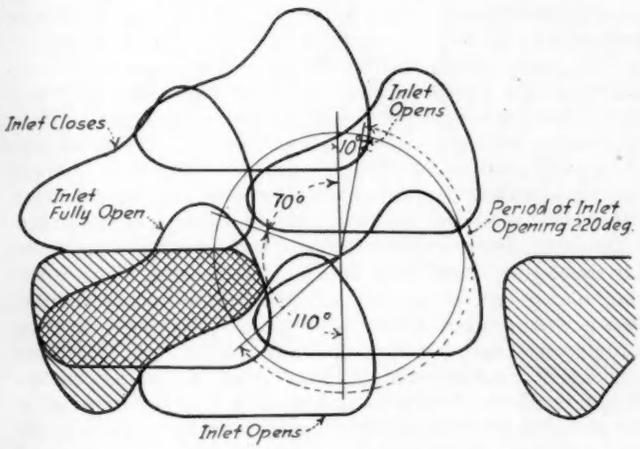

FIG. 10—THE BURT-TYPE INLET-PORT ACTION
The Cross-Sectioned Area Represents the Port in the Cylinder-Wall, While the Other Similar Areas Represent the Port in the Sleeve in Different Positions

times been questioned. However, at this time the pressure in the combustion-chamber is very small and the very excellent brake mean-effective pressures obtained at low speed are proof that there is no appreciable loss of charge at this period of the cycle if the engine is in good condition.

CIRCULATION THROUGH HEAD JACKETS

One thing that distinguishes sleeve-valve engines from other types is that the cylinder-head extends down into the cylinder a considerable distance. The cylinder-head is cast with a water-jacket that is open at the top but closed by a cover plate having water-tight joints with the central spark-plug well and the top surface of the cylinder casting. Usually, there is direct communication between the jacket around the cylinder and the head-jacket space; but, in one design at least, the two parts of the jacket communicate through outside pipes. The jacket space in the head forms a sort of well, and special provisions must be made to drain the water from it when the cooling system as a whole is drained. Generally a U-tube is provided, the two branches of which extend nearly to the bottom of the jacket space in the head and to a somewhat lower point in the jacket around the cylinder, respectively. If the water is being drained from the engine, it will siphon out of the cylinder-head jacket-space.

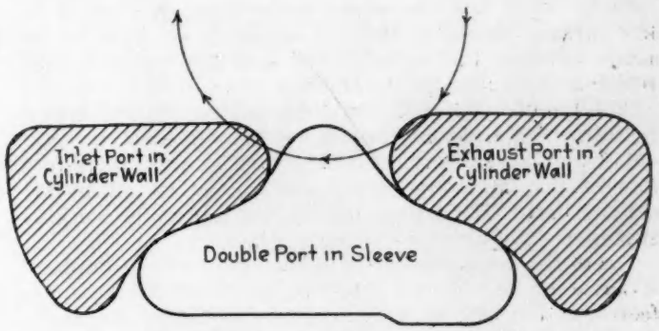

FIG. 11—DOUBLE PORT IN THE OSCILLATING SLEEVE
One of the Ports in the Sleeves Is a Double Port, Which Registers with an Exhaust-Port in the Cylinder-Wall during the Downward Motion of the Sleeve and with the Inlet-Port in the Cylinder-Wall During the Upward Motion

MERCEDES-KNIGHT ENGINES

The Mercedes Co. has developed three types of Knight engine in succession. The first one, which followed the original Knight practice in many respects, having front-end chain-drive and splash lubrication, was used up to the time of the war. After the war another design was brought out in which gear drive of the eccentric shaft was substituted for the chain drive and, to reduce fluctuations in the pitch-line velocity, the eccentric-shaft drive was placed at the middle of the engine which, like the previous one, had four cylinders. Another change in this model was that the accessories, that is, the pump and the magnets, were driven through spur gears directly from the eccentric shaft, instead of being driven by helical right-angle gears as in the original design. Placing the gear for the eccentric shaft between the two middle cylinders necessitated a considerable center distance between these two cylinders, and the cylinders were therefore cast in pairs.

The latest Mercedes-Knight engine, shown in Fig. 6, which has been in production since 1922, is a four-cylinder-block engine and has the upper portion, but not the upper half, of the crankcase cast integrally with the cylinders. The eccentric shaft in this model is driven by gears having helical teeth and located at the flywheel end. Aluminum-alloy pistons with cast-iron skirts are used. Lubrication is by the usual pressure system, the

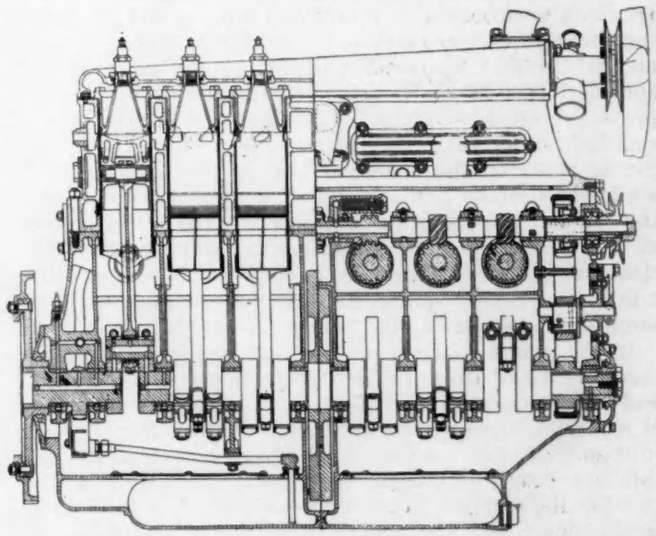

FIG. 12—VAUXHALL BURT-TYPE SINGLE-SLEEVE-VALVE ENGINE
This Is a Six-Cylinder Engine of Approximately 3¼-In. Bore and 5-In. Stroke and Is Shown in Longitudinal Section

cylinder-walls and the sleeve-walls being lubricated by the spray. However, there is an extra oil feed to the space between the cylinder-wall and the cylinder-head, which is controlled by the throttle valve.

The Daimler Motor Co., of England, the original manufacturer of Knight engines, in its latest models, brought out nearly a year ago, also uses steel sleeves, force-feed lubrication and a supplementary supply of oil to the cylinder-head. This company claims maximum working speeds of 4000 r.p.m. for stock engines.

One other improvement is to the credit of the Panhard & Levassor Co. In all other Knight engines the inner sleeve, at the bottom, extends below the outer sleeve, as the lug of the inner sleeve must at all times remain below the outer sleeve. This, of course, makes it necessary to add that much to the over-all height of the engine. Panhard obviates this by cutting a wide slot in the lower end of the outer sleeve, into which the lug on the inner sleeve moves. A narrower slot is cut in the inner sleeve, the object of which is evidently to provide clearance for the connecting-rod. The elimination of the wide junk-ring also has led to a shortening of the sleeves and of the over-all height of the engine.

Single-Sleeve Engines

The single-sleeve-valve engine referred to in the earlier part of the paper was invented by Peter Burt, a Scotch engineer. A single sleeve cannot perform the functions of both an inlet and an exhaust-valve if endowed with merely a simple reciprocating motion, and the Burt sleeve therefore has a combined reciprocating and oscillating motion imparted to it. This may sound complicated, but the valve is in fact fairly simple. Referring to Fig. 7, running parallel to the crankshaft in an upper compartment of the crankcase is a shaft that is driven from the crankshaft by gearing or silent chain, but at crankshaft speed instead of one-half that speed as in the conventional engine. This shaft carries helical pinions equal in number to the cylinders of the engine, each pinion meshing with a helical gear of twice its number of teeth, the axis of this gear extending transversely across the engine and intersecting the cylinder axis. The shaft of the gear carries a crank at its inner end and, on the crankpin, there is a spherical bearing which is supported in a lug formed on the lower end of the sleeve and is adapted to slide on the crankpin. Any point on the crankpin describes a true circular motion, but the bearing, owing to its connection to the sleeve, also slides in-and-out so that the motion of the center point of the spherical seat of the bearing is represented by a circle projected on a cylinder. However, the sleeve has a smaller radius than the distance from its axis to the center point of the spherical seat of the bearing and, while the bearing and the sleeve in their motion around the cylinder axis describe the same arc, the actual motion of the sleeve is less. For this reason the path of the sleeve as seen from the side is elliptical in shape. The 2 to 1 reduction is obtained for each sleeve separately by the pinion and gear at the bottom of the sleeve.

In the original design of the Burt engine there were three inlet and three exhaust-ports in the cylinder-wall, and five ports in the sleeve, one of the sleeve ports being of a form different from the others and serving for both inlet and exhaust. In some of the more recent designs this has been reduced to three ports for the inlet and two for the exhaust in the cylinder-wall, and four ports in the sleeve, but the general principle of operation remains the same. The outline of the ports also has been changed somewhat.

Form of Ports

The general form of the ports is not unlike that of a right-angle triangle, with the base up in the case of the cylinder-wall ports and down in the case of the sleeve ports. In one of the designs of Burt engine the throw of the crankpin is $\frac{3}{4}$ in.; hence, the stroke of the sleeve in the cylinder is $1\frac{1}{2}$ in. and its circumferential travel is a trifle less. The angular range of motion of the sleeve can be found readily from the diagram shown in Fig. 8. Let R be the distance of the center of the spherical bearing in the lug on the sleeve from the axis of the sleeve and r the radius of the crankpin operating the sleeve; then, if we call the angle of angular motion Θ, it is obvious from the diagram that $\sin \Theta/2$ equals r/R. For instance, if $r = \frac{3}{4}$ in. and $R = 2$ in., then sine $\Theta/2 = 0.375$, $\Theta/2 = 22$ deg. and $\Theta = 44$ deg. The piston diameter of this engine, and hence the bore of the valve sleeve, is $3\frac{1}{4}$ in.; hence, the circular motion of a point on the inside of the sleeve is $1\frac{5}{8} \times 6.28 \times 44/360 = 1\frac{1}{4}$ in., approximately, as compared with an up-and-down motion of $1\frac{1}{2}$ in.

The valve action can be studied to advantage by considering the ports formed in plane surfaces, of which one moves in a circle while the other is stationary. This, of course, involves equal circular and up-and-down motions, and that in reality the circular motion is less can be compensated for by changing the relation between the height and the width of the ports correspondingly.

To make the port action clear, I have drawn diagrams for both the inlet and the exhaust-ports. The ports shown, the valve timing and the angular relation between the engine crankshaft and the valve-operating cranks are not taken from any particular Burt-type engine, but correspond substantially with the practice of the builders of these engines. In Fig. 9, the total of the cross-hatched area represents the exhaust-port in the cylinder-wall, while the two unsectioned areas adjacent to it and the similar area midway between them and registering with the cross-sectioned area represent the exhaust-port in the sleeve in three different positions. In the highest position of this port it is about to come into registry with the cylinder port; that is, the exhaust period is about to begin. In the intermediate position of the sleeve port it is centrally over the cylinder port, and the exhaust is then approximately fully open. Whether or not this is the maximum opening can be determined only by a planimeter, which I did not have available. In its lowest position the sleeve port has just passed out of registry with the cylinder-wall port and the exhaust period therefore has come to an end. The motion of the sleeve port from the opening to the closing position, as shown in Fig. 9, represents 120 deg. of travel of the sleeve-operating crank; therefore, it also represents 240 deg. of travel of the crankshaft, which is substantially the duration of the exhaust period in any high-speed engine. Now, we can, of course, vary the angular relation of the sleeve-operating crank to the crankshaft and thereby vary the timing. Usually the exhaust-valve closes slightly after the crankshaft has passed the top dead-center, and we can assume that it is desired that it shall close at 10 deg. of crank motion after dead-center. In the diagram, Fig. 9, the sleeve-operating crank is shown to be in the bottom dead-center position when the exhaust-port closes, and this position of the sleeve-operating crank must then correspond to a position of the crankshaft 10 deg. past the top dead-center, as indicated in the diagram.

Attention is called to the rapid opening of the exhaust-

port. It will be noticed that the port in the sleeve moves with substantially its full width across the port in the cylinder-wall and, since the sleeve-operating crank is approaching the 90-deg. point, the motion is nearly straight down and therefore at nearly maximum speed. For this very reason it should not be necessary to give the exhaust a very large lead.

Inlet-Ports

A similar diagram of the inlet-port action is shown in Fig. 10. The cross-sectioned area again represents the port in the cylinder-wall, while the other similar areas represent the port in the sleeve in different positions. When the sleeve port is in the lowermost position, it comes into registry with the cylinder-wall port and the inlet period begins. Since the sleeve-operating crank is then in the lower dead-center position, this occurs when the crankshaft has passed the top dead-center position by 10 deg. The inlet period usually is somewhat shorter than the exhaust period, and the inlet-ports therefore are made of somewhat smaller vertical height than the exhaust-ports. In the diagram, Fig. 10, the inlet period is 110 deg. of sleeve-operating crank-motion and 220 deg. of crankshaft motion and, since the inlet begins to open at 10 deg. after top dead-center, it closes $(220 + 10) - 180 = 50$ deg. past bottom dead-center.

In the same manner as that in which the exhaust-port opens very rapidly, so the inlet-port closes very rapidly; in other words, the inlet-ports remain open comparatively wide until late in the inlet period, which should give a high volumetric efficiency.

In the diagram shown in Fig. 10, the position of one of the inlet-ports in the sleeve is shown at a number of nearly equal angular intervals throughout the cycle. The inlet is open during substantially the first half of the upward motion of the sleeve, while, during the remaining half of the upward motion and during the whole of the downward motion, the sleeve port is out of registry with the cylinder-wall port. From the path followed by the sleeve port the spacing of adjacent cylinder-wall ports can be determined for, during the last half of its downward motion, the sleeve port must pass between adjacent cylinder-wall ports without coming into registry with either one of them. A certain minimum margin between the sleeve port and the adjacent cylinder-wall ports is undoubtedly required, because the sleeve port approaches most closely to the cylinder-wall ports during the early part of the exhaust stroke and, if the seal of the inlet-ports were poor at this time, some of the exhaust gas might get into the inlet passage and dilute the charge for the next cycle. Just how much this margin should be is obviously one of the fine points of single-sleeve-valve design. By making this margin smaller, the total port-opening area can be increased.

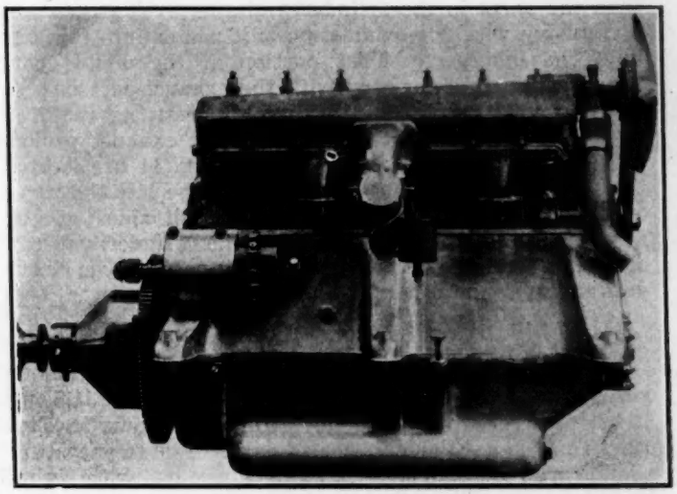

Fig. 14—Manifold Side of Vauxhall Burt-Type Engine
Located between the Halves of the Middle Bearing Is the Oil Thrower for the Toothed-Chain Drive for the Magneto and Generator, and an Eccentric for the Oil-Pump Drive Is Mounted between the Halves of the Rear Bearing

Double Port in Sleeve

One of the ports in the sleeves is a double port, which registers with an exhaust-port in the cylinder-wall during the downward motion of the sleeve and with the inlet-port in the cylinder-wall during the upward motion. The arrangement is somewhat as shown in Fig. 11. The double port is of considerably greater width

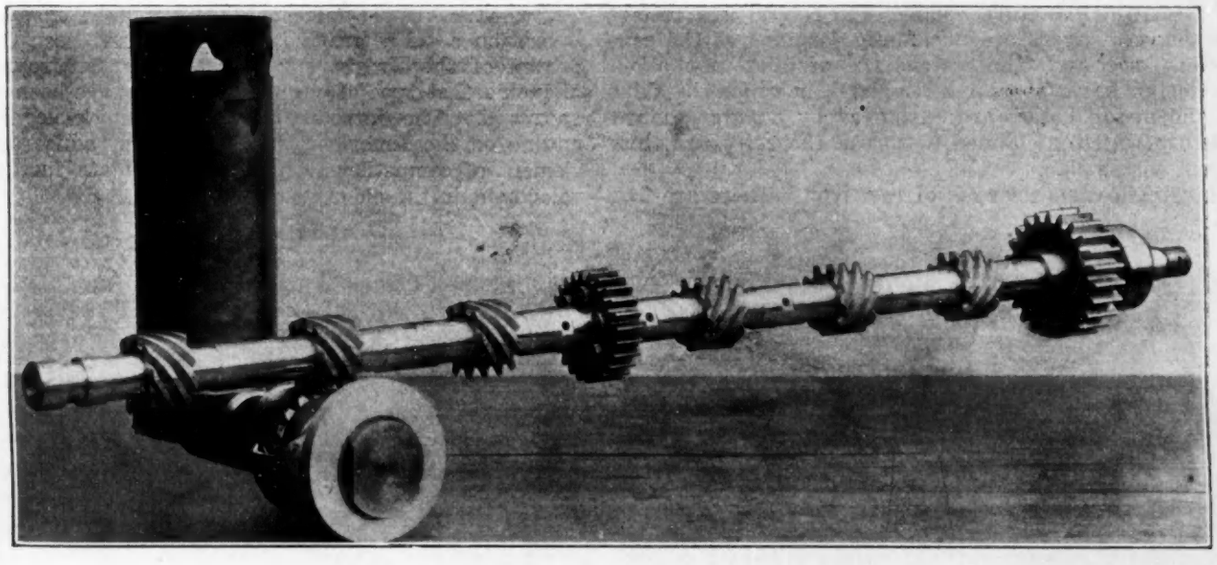

Fig. 13—Vauxhall Burt-Type Sleeve-Operating Shaft
To Balance the End-Thrust on the Sleeve-Operating Shaft, Three of the Helical Gears on It Are Cut with Right-Hand and the Other Three with Left-Hand Helical Teeth

than the ports in the cylinder-wall and, as it passes out of registry with the exhaust-port, it passes into registry with the inlet-port. That portion of the double port which comes into registry with the exhaust-port only is of somewhat greater depth than the rest, as required by the greater angular duration of the exhaust period.

To the right of the double port in Fig. 11 are located the two or three exhaust-ports, and to the left the three or four inlet-ports. The backs or high sides of the farthest inlet and exhaust-ports in the cylinder-wall can come fairly close together, because the ports in the sleeve do not travel materially beyond these lines.

Vauxhall Single-Sleeve Engine

One of the latest Burt-type engines to be brought out abroad is a six-cylinder Vauxhall engine of 81.5 by 124.0 mm. (nearly 3¼ x 5 in.), which is shown in Fig. 12. To balance the end-thrust on the sleeve-operating shaft, three of the helical gears on it are cut with right-hand and the other three with left-hand helical-teeth. The sleeve-operating shaft shown in Fig. 13 is driven from the crankshaft through a toothed chain, and the timing of the lay-shaft can be effected accurately by a vernier arrangement. There are 10 main bearings on the crankshaft, which is due to the fact that the front, middle and rear bearings are each split in two, the sprocket wheel for the timing-shaft drive-chain being located between the halves of the front bearing. What looks like a flywheel, but which serves as an oil thrower for the toothed-chain drive for the magneto and generator, is located between the halves of the middle bearing, as shown in Fig. 14, and an eccentric for the oil-pump drive is mounted between the halves of the rear bearing.

The crankshaft is of rather unusual construction, being built-up of units which are clamped together. The crank-arms are pressed over the main-journal portions, and the arms are split and V-notched at their outer ends and have the separate, hardened crankpins pressed into them and held firmly in them by pinch bolts.

Floating bronze-bushings are fitted between the connecting-rod heads and the hardened-steel crankpins and are provided with numerous oil-holes. The advantage claimed for this construction is that it permits narrower crankpin-bearings to be used than would otherwise be the case, which in turn reduces the centrifugal forces. The connecting-rods are duralumin forgings of the conventional section. The pistons are of aluminum alloy, of a design fairly popular in England, in which the skirt and the piston bosses are supported by an arch-shaped structure which is claimed to spread the skirt and thus prevent piston-slap.

The single valve-sleeve is of cast iron. There are five ports in the cylinder-wall, of which three serve for the inlet and two for exhaust. In the valve sleeve there are four ports, of which one is a duplex port.

The combustion-chamber is of the general form of a truncated cone. It will be observed from the sectional view in Fig. 12 that it is well jacketed and, as the form of the piston also suggests the possibility of a rapid transfer of heat to the cylinder-jacket, it is understandable that an unusually high compression can be carried The volumetric compression-ratio is 5 to 1, which is probably higher than has been carried before in a passenger-car engine.

The cylinder-head extends down into the cylinder and forms an annular space into which the upper end of the valve sleeve extends, the same as with the double-sleeve-valve engine. Owing to the fact that the ports are comparatively narrow in the circumferential direction no wide ring is required. In fact, only a single ring of normal width is used.

As regards the balance of a single-sleeve-valve engine, what was said in connection with the double-sleeve-valve engine applies in this case also; that is, in a four-cylinder engine, the primary unbalanced forces on the sleeves will not be neutralized, but the secondary forces will be neutralized. Of course, with two sleeves, of which one leads the other by say 70 deg., the total unbalanced force on the sleeve is greater by about 64 per cent than on a single sleeve of the same weight as each of the two.

In the single-sleeve-valve engine we have an oscillating motion of the sleeves in addition to the reciprocating motions, but these oscillating motions are at least very nearly balanced.

No comparison of the merits of the different types of engine will be attempted in this paper. In judging these merits one would need to take account of at least six factors; namely, cost of production, weight efficiency, fuel efficiency, reliability, life, and freedom from noise in operation. It is hardly conceivable that any one type could excel in respect to all these factors and, moreover, since it is practically impossible to weigh superiority in one of these items against superiority in another, any conclusion based solely on theoretical considerations would be of no value.

Just as competition is the life of trade, so is competition a stimulant to technical progress. It is very doubtful whether poppet-valve engines would be as silent in operation as they are today if it had not been for the advent of the Knight engine; at any rate, progress in silencing that type of engine would not have been nearly so rapid. The introduction of the single-sleeve-valve engine on the American market may introduce another element of competition which will stimulate further development in the older types.

Discussion of Papers at 1925 Production Meeting

THE discussion following the presentation of five of the papers at the Production Meeting of the Society held at Cleveland last September is printed herewith. The authors were afforded an opportunity to submit written replies to points made in the discussion of their papers and the various discussers were given an opportunity to approve the stenographic report of their remarks before publication, as is required by the Rules of the Society.

For the convenience of the members brief abstracts of the papers precede the discussion in each case so that some idea of the subjects covered can be obtained readily. Members who desire to refer to the complete text as originally printed will find all five papers in the October, 1925, issue of THE JOURNAL. Since the publication of the paper by Franklin T. Jones, entitled Products and By-Products of Foremen's Conferences, in the January, 1926, issue of THE JOURNAL, with such discussion as had been approved for publication up to the time that issue went to press, an additional discussion by R. D. Bundy has been received. This discussion is also printed below, and is preceded by a brief abstract of the paper. The discussion following the presentation of the paper by W. G. Careins, entitled How To Select Machine-Tools, will be found, together with the paper, in this issue of THE JOURNAL, commencing on p. 275.

THE PROBLEM OF GEAR PRODUCTION

BY EARLE BUCKINGHAM[1]

ABSTRACT

STATING that the production of gears presents problems more baffling than those of other lines of production, the author believes the first essential for satisfactory results to be suitable means for detecting and measuring errors. The main difficulty here lies in devising means of measurement that are at once simple, rapid and effective, rather than to rely upon delicate laboratory-instruments unsuited for usage in production work.

Very slight errors in gears cause serious troubles and, since gears must be hardened, their production demands a high degree of precision under the most unfavorable conditions. Forming and generating constitute the two general classes of machining. Generating processes are of two types: in the first, the generating tool represents the form of the basic rack of the gear system and, in the second, the tool represents one of the gears of the system. A burnishing process also has been developed which smooths the profiles of machined gears by crowding them into accurate hardened-and-ground burnishing-gears. This process will make a good gear better, but it will not make a bad gear good.

Refinement of methods and reduction of cost are discussed, it being emphasized that the variable factor due to the differing human equations of the workmen who operate the production equipment is the greatest difficulty of any production problem.

THE DISCUSSION

R. S. DRUMMOND[2]:—Our company has had some rather interesting experience in the development of machines for measuring gear teeth during the last 5 years. We bought nearly every type of machine that we could secure, looked it over and tried to use it in our shop. As a result of that experience, gained by operation of the measuring machines, we felt that we should go a step farther. We developed a machine that has a disc mounted on a horizontal bar and an arbor to hold the gear and index it. As the gear moves 1 deg. at a time, it moves the bar as a rack. Another parallel bar is controlled by an accurate micrometer. A comparison can be made of the movement of the bars for each degree of angular movement of the gear under inspection. Some of the results obtained in testing gears in that way have been rather notable.

This machine has been used not only on spur-gears but on a number of helical timing-gears, spiral gears, bevel-gears, and spiral-bevel gears. The development of its usefulness has not been 100 per cent, but it has helped considerably. This machine is useful for laboratory investigations and in setting machines. In production a machine such as I first described can be used for every-day work if 1 piece out of every 100 is measured.

The comparing device, which we also use, is more rapid and, therefore, more desirable for production purposes. It has a vertical arbor and carries a master gear in contact with a rack-bar. The production gear on the same arbor is in contact with a similar parallel rack-bar. By revolving the gear arbor, the rack-bars are moved simultaneously and direct comparison is obtainable between a tooth on the master gear and one on the production gear. This test occupies only 15 sec. per gear. It is possible to check one gear for every arbor of gears that is being run. We do not let our grinding machines run until a truck load of gears has been accumulated but as soon as an arbor of gears is ground they are checked. At the rate of one gear per 15 sec. one man can handle a large number of gears.

CHAIRMAN H. W. ALDEN[3]:—I am surprised that more has not been said about the accurate mounting of a pair of these fine gears after they have been made. I assume that Mr. Tenney has discovered that no small amount of wear, pitting and noise come from not putting the two gears together the way they were intended to run. A transmission builder labors under the disadvantage that he cannot fit the gears closer together to transmit the load. Consequently, he must have a certain amount of spread and deflection of the shafts. I know that if we had to mount bevel-gears with bearings on the shaft as far apart as they are on the transmission shaft, no cars would be built unless the public would accept something they did not want. With regard to the accuracy of ma-

[1] M.S.A.E.—Engineer, Niles-Bement-Pond Co., New York City.
[2] M.S.A.E.—Vice-president, Gear Grinding Machine Co., Detroit.
[3] M.S.A.E.—Chairman of the board of directors, Timken-Detroit Axle Co., Detroit.

chining the part that the gears are put into, I assume that a company which makes its own gears and transmission cases watches that carefully, but I am always sorry for the man who has to make gears for someone else's case.

Bearing in mind all of this good brain work that is put into the making of a train of gears 100 per cent perfect and getting the tooth contour down to ten-thousandths of an inch and so on, I do not believe enough attention is paid to the place where the gears are put to work. From our own experience I think I can say that one of the most important considerations, if not the most important, with regard to bevel-gears, is to put a pair of gears to work just where it was intended to be put and to keep it there. If that is done, half of the troubles will be over. I rather suspect that failure to do that is one of the causes of the troubles with transmission gears.

O. GARRISON[4]:—While on the subject of simplicity of inspection devices, I should like to know what has been the conclusion regarding the roll-test. We have found in the roll-test that, regardless of whether the rolling is done with the master gear or one that has been cut or ground previously, a fluctuation of the indicator attached to the sliding member of the testing device will indicate a number of errors. It will give a reading as to eccentricity, errors in tooth spacing and tooth forms, and it has been our experience, whenever we get a fluctuation, that we go to the involute and tooth-spacing tester to find out where the difficulty is. What is the objection to the use of that method of testing?

MR. DRUMMOND:—The test referred to, which is commonly known as the "jiggling" test, consists in putting two gears together and revolving them while they are held in solid contact by spring tension. The vibration of the indicator needle shows that something is wrong. Anything from a piece of dirt to the pressure that is put on the gears will produce a jiggle of the indicator. I do not have much confidence in the ability of the jiggling test to help in making good gears. We can find out that something is wrong in a number of ways but that does not solve any of the trouble.

In our testing we eliminate the jiggling test and use various devices for measuring the larger fundamental errors in the gears. The eccentricity test is good and the inspection of the index error is another good test.

Considerable is heard about tooth-form measuring and not much about the other tests. Tooth-form measuring is one of the most important tests in the control of gear-making.

QUESTION:—Would it not be well to attempt to eliminate the term "spiral gear" when really a helical gear or skew gear is meant? This would eliminate the ambiguous term "spiral."

EARLE BUCKINGHAM:—"Spiral gear" is a misnomer, except for bevel-gears. The tooth is a spiral there. The main reason for the use of the term is that the term "helical gear" is used for the gears that drive on parallel axes while "spiral gear" is used for gears that drive shafts at an angle because of the lack of any other term. Technically, the use of the term for such gears should be discontinued.

G. L. MCCAIN[5]:—Some mention has been made of long-addendum-tooth gears. I was connected with engineering work on bevel and long-addendum gears and also with the production of gears for the Liberty-12 engines, in which every ounce of material had to be made to work and in which the long-addendum gear, although noisy, was used because of its strength. I should like to hear from someone regarding the difficulty in cutting such gears. I believe it is immaterial whether the gears are cut or ground when the radical difference in tooth form is considered and when the long-addendum tooth and circular and diametric pitch are odd and are made to conform with the center distance. It is the tooth combination that must be considered.

MR. BUCKINGHAM:—The difficulty in cutting such gears is more mental than actual. It is necessary to do a little harder thinking. Of course, if the tooth is deep, more material must be removed. The problem is like all others, possessing some advantages and some disadvantages, some increased difficulties and some improved conditions.

MACHINE-TOOL NEEDS OF THE AUTOMOTIVE INDUSTRY

BY R. M. HIDEY[6]

ABSTRACT

RAPID automotive-engineering developments and keen competition, fostered by public demand for better and cheaper transportation, have provided the impetus for most of the development in the machine-tool industry for the last 15 years, and the author analyzes present needs for machine-tools as a basis for his recommendations for improved methods of business procedure and better cooperation.

After discussing the relative merits of single and multi-purpose machine-tools, matters relating to the safeguarding of such tools, the benefits attendant upon better cooperative effort and the necessity of making greater expenditures for the development of the machine-tool industry, the author advocates active pioneering in fresh fields that offer greater opportunities for machine-tool usage, rather than dependance upon replacement business, and makes five specific recommendations to machine-tool builders.

THE DISCUSSION

A. C. COOK[7]:—Mr. Hidey's paper confirms several views that our company has held for a long time. One of these is that the machine-tool industry learns from its customers; that is, it must develop machine-tools for certain needs in the field of consumed goods. A machine may be developed for certain work but the machine-tool builder has not, of course, as much opportunity for developing the use of the machine and its productive capabilities as the customer who buys the machine and puts it to work in his plant. We acknowledge indebtedness to the automotive industry in this connection, for a number of the important features on different types of machine-tool have been placed there as a result of the things we have learned from that industry.

The building of special machines for use in the auto-

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[5] M.S.A.E.—Automotive engineer, Link-Belt Co., Detroit.

[6] Works manager, White Motor Co., Cleveland.

[7] Vice-president, Warner & Swasey Co., Cleveland.

motive industry involves exactly the same problems that a passenger-car builder would have to solve if he attempted to design and build half a dozen 10-ton trucks in a plant that is equipped to produce passenger cars.

The fields for the passenger car and for the truck are distinct, but the service required from each type of vehicle is different. The same thing applies to machine-tools, in that the standard machine-tool of any type can be produced in quantity and thus a quality article can be produced on a quantity basis.

Standard Machines More Profitable to All

The fact that the automotive manufacturing plants are equipped today largely, I might say almost entirely, with standard machine-tools, supplemented, however, with such special tools and fixtures as may be required for certain parts, amply bears out Mr. Hidey's statement that the industry has found the use of such standard machines more profitable than the use of special machines. In the first place, the standard machine-tool, being produced in quantity, can be bought at a more reasonable price than the special machine, and, in the second place, it is a machine with which the ordinary workman is familiar. The experience of the machine-tool industry parallels that of the automotive industry in that single-purpose machines are economical only when used on production in immense quantities.

Standard machines are the bread and butter of the machine-tool industry. It might be thought that the special tools and fixtures that are furnished to the automotive industry are the cake, or even the cake and ice cream, but if the cost records on the making of these special tools and fixtures could be seen, probably you would wonder how we have any cake and ice cream at all. As a matter of fact, the profit on them is very small.

As Mr. Hidey says, a real demand exists for multi-purpose machines in all metal-working industries. The multi-purpose machine is flexible, simple in construction and very easily and quickly changed over from one job to another as production schedules require, or as may be required by the development of new car models and changes in design. The intent of the machine-tool industry is to give the greatest possible value for every dollar of cost.

Conditions That Impede Cooperation

Mr. Hidey mentioned the lack of cooperation among the builders of various types of machine-tool. It is true that cooperation seems to be lacking but the opportunity for it in our industry is not so great as in the automotive field, owing to the nature of the industry itself. The automotive industry manufactures one product in two forms, that is, a gasoline-propelled vehicle, whether it be a passenger car or a truck. Certain problems are identical with all manufacturers, for all use the same kind of units although these vary in size, shape and otherwise. All motor vehicles have similar unit members, such as engine, transmission, differential, four wheels, and a body. The same parts can be used by a number of different manufacturers and when the parts are assembled the result is a motor vehicle. The machine-tool industry cannot do this, for the requirements of customers throughout its entire field cannot be served by one type of machine. This field, of course, is metal-working, but metal must be turned, bored, drilled, planed, or milled, and each of these operations presents such a different problem that each process must be performed by machines designed for that particular part of the industry. For example, the operations most commonly performed on engine lathes are turning, threading and facing, and the machine cannot well be used as a milling-machine or a vertical drilling-machine, hence the problems in the engine-lathe field are peculiar to that field, as are the problems in the milling-machine field, which can be solved best by the milling-machine builders.

Members of the National Machine Tool Builders Association cooperate in various matters but obviously the companies in the several groups represented in the Association must cooperate among themselves to obtain the best results as applied to their own particular types of machine. Thus, the vertical drilling-machine builders cooperate as a group and so do the milling-machine builders. The only problem that is common to all of the groups is that of removing metal, and information and experience along these lines is interchanged freely among the various groups.

Constantly at Work on Design Changes

The problems of the machine-tool industry with regard to designing and experimenting are constantly in the process of solution. Those relating particularly to designing are very different from the problems of the automotive industry, due more than anything else to the fact that we do not manufacture a consumers' article. Therefore we do not need to depend upon the whims of the consuming public. When a customer specifies some particular type of machine-tool of a certain make he does not do so because the machine is painted a certain color or is finished in a certain way. He tries to select the machine that will earn the most money and give him the greatest return on his investment. This accounts for the fact that changes in the design of machine-tools are not made more frequently, although such changes are made as often as conditions seem to demand, for instance, a number of years ago the entire industry was practically revolutionized because high-speed steels came into prominence suddenly, thus rendering almost obsolete all of the knowledge we had gained from the use of carbon-steel and requiring stronger machine-tools to take advantage of increased cutting-speeds and feeds. This process of improvement in design has been going on constantly, particularly during the last five years when our industry had plenty of time for study. This time was put to good use in programs of redesigning throughout the industry.

Certain machine-tool builders may expect their customers to do their experimenting for them but I am inclined to think that the majority experiment long and carefully with a new product before it is placed on the market. I know of one case in which a machine of a new design that was a refinement of an earlier design was tested and experimented with by the builder for nearly two years before being offered to the trade. The fact that the machine has gone into service with very few subsequent complaints shows that the builder did a very good job.

Standardization a Knotty Problem

The standardization of machine-tools is a rather knotty problem. I am glad to say, however, that the National Machine Tool Builders Association has a committee working actively on this subject in connection with the American Society of Mechanical Engineers. This question has several sides, and one of them that occurs to me may be illustrated thus: Suppose that you have in your shop 50 engine lathes of the same size and make and with the same spindle-thread, and suppose further that the machine-tool industry decided to adopt as a standard for

16-in. engine lathes a different thread from those you have on your machines. You can well imagine the confusion that would result. If the spindle-threads in all 16-in. lathes were to be made identical, it is probable that no existing spindle-thread would be used and the only way to solve the problem would be to compromise on a non-existant thread.

In certain lines of machine-tools special spindle-threads can be made to order. Thus, if the 50 engine lathes now in your shop are equipped with the thread of one builder and you wish to purchase some 16-in. lathes made by another company, I am sure the latter would be glad to equip his machine with the thread you are using now, which would solve your problem. This is done frequently and probably causes less confusion than if all the manufacturers were to change over and use the same threads and other details.

Tool Association Adopted Standard Data-Sheets

Our industry recognizes the value of standard data-sheets to be filed in engineering offices for reference when it is necessary to lay out new tools and jigs. The National Machine Tool Builders Association some time ago adopted as a standard a data-sheet prepared by A. L. Evans and founded upon his experience as a tool designer in a large automotive plant. All of these sheets are of identical size so that they can be placed in the same binder, and each sheet shows the principal dimensions of the particular machine for which it is prepared, so that users of machine-tools can have in their records the necessary data to enable them to ascertain quickly whatever dimensions are required without measuring a machine or writing to the builder for the information. A number of machine-tool builders now send these data-sheets with each machine that is shipped.

More Safety Devices Being Fitted

Safety devices are being built into machine-tools more and more. Some of these are very simple because not many parts in the standard machine-tool need to be guarded. On single-pulley-drive machines, such as engine lathes, where it is not necessary to shift the driving belt, belt guards are attached, and on some machines the entire pulley is guarded. On cone-drive machines danger arises of the operator getting his hand caught between the pulley and the belt when trying to shift the belt. Obviously, a belt-shifter is the proper means for avoiding this, but it is almost impossible to build one into the machine. They are very simple in construction, and are easily attached in the shop. The safety problem is, of course, more complicated on punch-presses and other tools of that nature, but the builders are constantly endeavoring to render the machines accident-proof.

At a meeting of the American Society of Mechanical Engineers held some time ago, a session was devoted to the question, "What has the machine-tool industry learned from the automotive industry?" It has learned considerable from your industry but I think the one big thing we have learned is the broader spirit of cooperation that has developed during the years that our two industries have been working together.

Chairman Eugene Bouton[8]:—Does the machine-tool builder supply data sheets for obsolete machines upon request?

Mr. Cook:—I do not know that the request has ever come to us, but we could prepare such a data sheet.

Chairman Bouton:—The reason I ask is that the production men in general do not know that these data sheets have been adopted.

Mr. Cook:—We can make up the data sheet for any of our machines.

Buyers Try To Beat Prices Down

E. F. DuBrul[9]:—I think that while Mr. Hidey's paper is very good it draws a number of general conclusions from a limited number of examples and that is bad logic. In his eagerness to secure better machine-tool service he makes it appear that all machine-tool builders are at fault. I think he could find a number of exceptions to this conclusion.

To hear the tales of the salesmen who sell machine-tools to some of the "hard-boiled" buyers in the automotive shops always amuses me. A big difference exists between markets and outputs of machine-tool builders and manufacturers of automotive equipment. The latter are used to producing parts in large numbers, which naturally reduces the unit cost. The automotive buyer knows his own costs for labor and material pretty well, but knows little of machine-tool costs. When the machine-tool salesman enters his office the buyer argues illogically from his premises against the price of machine-tools produced under entirely different conditions.

Producing any kind of machine-tool in large numbers is impossible because machine-tools of any given type are not bought in large numbers. If a builder produced them on a large-number basis he would go into bankruptcy before they could be sold. A considerable number of men have gone from the machine-tool industry into the automotive industry but I have not noticed any great rush of automotive men going into the machine-tool industry to make a fortune. Perhaps there is a reason.

Automotive Industry Does Not Pay Enough for Machines

A specific point raised in Mr. Hidey's paper was that proper care is not taken of the requirements of the automotive industry. It may be that the automotive industry has not been willing to pay enough to be taken care of under the conditions of cost that rule in the machine-tool industry.

Mr. Hidey quoted me as saying that the machine-tool industry depends upon the replacement business rather than upon new industrial developments. I feel sure that I did not make that bald statement without qualifications. The fact is very much the contrary. To make a machine-tool that is any good at all, it must be made so very good that the replacement demand due to wearing-out is very little. The industry cannot live on such replacements. Unlike an automobile a machine-tool is not worn-out in 3 or 4 years. The wearing-out is so slow that if we depended upon the replacement of worn-out machines we should starve to death. We have to cudgel our brains to bring out new designs that are more highly productive and to sell them to replace obsolete machines.

Do Special Machines Receive Too Much Attention?

The question of single-purpose machines is purely economic. If production is large enough to justify the purchase of a special machine for a certain operation and the saving will pay for the machine within an economic time-limit, that machine will be bought. The buyer may not be a good guesser and may not know how long the particular piece will be in production, but that is not the machine-tool builder's side of the equation. The line

[8] M.S.A.E.—Supervisor of time study, Chandler Motor Car Co., Cleveland.

[9] General manager, National Machine Tool Builders Association, Cincinnati.

between special and general-purpose machine-tools is not to be drawn by the builder but by the user. Demand is the mistress to whose whims every business must cater, and she is a very fickle mistress sometimes. The machine-tool industry must do what every other industry does, that is, supply what its customers want. If the customers are willing to pay for a special machine, they will find some one to build just what they want. Demand comes first and supply follows. Consequently one of the generalities which Mr. Hidey should modify is that too much attention is paid by the builders to single-purpose machines. As a matter of fact, few single-purpose machines are being built and most of the builders are desirous of selling general-purpose machines, of which they can build more than one machine at a time. However, if the user wants the single-purpose machine and pays enough for it, the machine-tool industry will build it.

State Laws Vary on Safety Guards

As to safety guards, the American Engineering Standards Committee's Sectional Committee on Safety Code for Machine-Tools is working on the problem. Our Association is represented on a sub-committee that is working on the correlation of safety codes of all of the different States. Throughout the United States, and everywhere else in the world, the codes are different. Every political subdivision has its own ideas of safety devices for the proper guarding of machinery. I tried to secure a digest of the safety codes of the entire United States for our members but nobody could furnish it. The Department of Labor at the City of Washington has no such digest, and the correlating sub-committee has none. The American Engineering Standards Committee will have to compile one.

Our industry cannot supply safety guards that will meet the requirements of all of the different laws of the United States, because safety guards that are required in one State are forbidden in another. The machine-tool builders cannot know what the varying requirements are in each State and have no chance to find out. Hence the blame for leaving off safety guards should not be placed on the machine-tool industry but upon the labor organizations, the politicians and others, who are responsible for the diverse practices of different States.

Something was said regarding machine-tool shops not being open to inspection by competitors. I do not know of a single shop to which its competitors are denied access, except to the research, engineering or designing departments. Mr. Hidey is mistaken in his impression that any secrecy exists as to methods. For years before the automotive industry existed, the machine-tool shops were as open as the automotive shops are today.

Cost of Development Overlooked

I am glad that Mr. Hidey takes the same stand that I do in the matter of cooperation between the automotive industry and the machine-tool builders, but the excessive cost of research and experiment in the machine-tool industry is overlooked by him. The builders are spending large amounts of money in research because they do not cooperate. I have observed many cases of unnecessary duplication of research work, and research work is expensive.

I know of no business in which the users of goods escape paying all of the costs and wastes of production. I do not know how the user can get his goods unless he does pay the cost plus a profit, because it is a fact that a profit must be earned or a business will not continue long and no goods will be made.

I am a firm believer in eliminating waste and reducing costs as much as possible, but it must not be thought that the machine-tool user can escape the burden of testing. I think Mr. Hidey has given a poor illustration and drawn a general indictment against a whole industry from an experience with one manufacturer who did not live up to his guarantee. To conclude from this that other builders all do the same is poor logic. He was merely unfortunate in choosing a builder who was over-enthusiastic in his estimate or something worse than that. Many buyers are careless and do not insist on compliance with contracts. Then again some men deliberately make guarantees that they know they cannot fulfill. The careless buyer, who should know better, accepts these estimates. When he receives the machine and finds that it will not do what it was guaranteed to do, he keeps it because the machine is about as good as others in the shop. That is not fair to the conscientious seller. That kind of business is not good for anybody, and the practice should be stopped. If the buyer lets the seller off in such a transaction, he should blame himself rather than the whole machine-tool industry.

Buyers Cause Lack of Standardization

I believe in standards. That the vertical drilling-machine group of our Association lists more different sizes and varieties of machine than there are Heinz pickles is ridiculous. Fortunately they are not all made by any one company. The buyers are responsible for the situation. They can have standardization when they want it. Demand for standardization will create standards. This ridiculous situation in excess varieties exists because some user thinks he wants a 21-in. swing instead of a 20 or 22-in. swing. Perhaps the user asks a builder to raise the head and tail-stock of a 16-in. lathe and make the swing 20 in. It would result in considerable benefit to the builders and the users of machine-tools if some elements of machine-tools were standardized. But, so long as the buyers insist upon having something special and a profit can be realized in making it, that is the way it will be made.

We have this same problem with the electrical industry. The men in that industry give many reasons why they cannot standardize certain dimensions of electric motors but our industry will get standardized motors when it wants them badly enough to insist upon getting them. Let the automotive industry make a concerted demand for standardization, and it will be forthcoming.

Mr. Hidey has rendered a real service in raising some of these questions. More organized cooperation should exist among various groups in the machine-tool industry as well as between the machine-tool builder and his customers.

Too Little Interchangeability

Chairman Bouton:—I took particular interest in what Mr. DuBrul had to say regarding salesmen's reports of interviews with the "hard-boiled" production men and speaking for production men I would like to say that very few machine-tool salesmen have a good working knowledge of the tools that they are selling and in fact cannot tell you very much in an interview. The machine-tool builders could do well to standardize on salesmen as well as on their machine-tools themselves. Production men generally are very anxious and glad to meet machine-tool representatives who thoroughly understand the product that they are selling but are just as anxious to evade the golf-stick type who is unable to describe thoroughly and talk understandingly about the machine that he is selling.

With reference to spindle-noses that Mr. Cook mentioned, a noticeable improvement has been made in standardizing the pitch diameters of the threads. Only recently has it been possible to interchange a chuck adapter from one machine to another; in other words in a plant having 10 machines all of the same size and make, it is rarely possible to interchange the adapters. A noticeable improvement in this respect has been found by many production men.

ERIK OBERG[10]:—Mr. Hidey has understood Mr. DuBrul's statement in one of his bulletins, that the machine tool industry depends upon replacement business rather than upon new-development business, entirely differently from the meaning that Mr. DuBrul conveyed to me and the meaning that I know he intended to convey. Mr. DuBrul meant that the machine-tool industry does not depend upon sales to new plants or new industries being started but has to depend largely upon replacements of equipment in existing plants. That does not mean replacement by the same old types of machine but the replacement of old machines in existing plants by new designs. A machine that is 10 or 20 years old cannot be replaced with a machine of the same design because very few such machines are available. I am sure that is the idea Mr. DuBrul meant to convey.

SHOULD DEMONSTRATE GUARANTEED PRODUCTION

W. W. NICHOLS[11]:—I have been much interested in Mr. Hidey's paper, especially in what he said about the testing of the machine being thrown upon the customer and also in his remarks about sending samples of various parts to different manufacturers.

In 1900, Alfred Herbert & Co. adopted the policy of making their quotations and giving guarantee-times on the production of their turret lathes or full automatics when drawings or samples of parts were submitted to them. If the quotation and the production rate were satisfactory to the customer, they were willing to build and equip the machine and put it into operation, using the customer's own material in his own shop, and demonstrate it fully. I never have known of a case in which the machine has failed in performance. I think very much of the trouble could be avoided if the customer would specify, when taking up the question of equipment with the machine-tool builders that this should be done.

I think it would also be of advantage if customers would permit the designers of the machine-tools to visit their factories to see the condition of their tools and under what conditions the tools operate, especially in the automobile plants of today. If that were done a material strengthening of the machine-tool parts and a reduction of the maintenance cost would result.

COOPERATION WOULD REDUCE COSTS

R. M. HIDEY:—What I endeavored to do in this paper has been accomplished. It has brought out a good discussion. I will agree with Mr. DuBrul that some of my remarks were generalities. Regarding closed shops, we hear that they are closed from the representatives of the machine-tool builders and men who come from the machine-tool industry to work with us. If the cooperative spirit existed more in the machine-tool-building industry, the experimental work, guards, installation, and so on would cost less. The automotive industry has been standardizing for the last 20 years. About 600 standards are now in practice and much more remains to be done along this line. The machine-tool builders have a similar opportunity in their field.

Forty-six machine-tool builders have adopted the data sheets referred to by Mr. DuBrul. The larger companies are using them. We have not tried to get data sheets for old machines but have drawn them ourselves. They result in a remarkable saving for the tool designer. How long would it take the machine-tool industry to give us something that would enable us to interchange the chucks on lathes and milling-machines?

Before introducing a bonus system in our toolroom recently, we made a check of all our lathes and found that all the lead-screws and the spindle-noses were different. We had to move some of the machines out and put in others so that we could operate our bonus system.

I think we all realize that considerable work remains to be done in the matter of safety guards. It is like the glaring-headlight problem in the automotive industry. It is still being studied.

AN ANALYSIS OF MACHINE-TOOL MAINTENANCE

BY A. R. KELSO[12]

ABSTRACT

MACHINE-REPAIR analysis and a criticism of present-day equipment, with analytical tables based on data collected from a 5 months' study, are followed by conclusions relative to the reliability of present-day equipment. Eight types of common machine-tools are considered and the maintenance advantage of one over the other is deduced from the consolidated tables based on monthly reports. A comparison of each class of machine-tools with the others is made, as well as a summary of the weaknesses of each class from the frequency of repairs of the elementary parts. The attention of the builders is drawn to the conditions that the shop encounters with their equipment.

A maintenance budget-system is described that has been installed in one plant to give the men a comparative idea of particular equipment that is running in excess of the budget time. It also serves in lieu of an inspection of the conditions of the equipment and is an indication of the time when overhauling is advisable. The analysis shows that a few makes of machine predominate in each group which can serve profitably as a basis for standardization. The other makes, in comparison, are not giving the same return on the investment, because of the expense charges and the loss of time for maintenance.

THE DISCUSSION

E. F. DUBRUL[13]:—Mr. Kelso's paper shows that somebody is analyzing machine-tool equipment and gathering statistics on maintenance instead of depending on guesses. When facts of that sort are collected, something is available to put one's finger on, and that is good engineering work. With that kind of information at his disposal, Mr. Kelso is able to do what I have often said would be a good thing for someone to do; that is to score or rate

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[11] A.S.A.E.—Mechanical engineer, D. P. Brown & Co., Detroit.

[12] Works manager, Continental Motors Corporation, Muskegon, Mich.

[13] General manager, National Machine Tool Builders Association, Cincinnati.

the machines according to performance so that the buyer would know the advantages and deficiencies of each machine. Too many purchases of machine-tools are made by intuition, guesswork, prejudice, favoritism, and general impressions without having any engineering basis for the decisions.

One of the problems in marketing a good machine is this; when a man has studied his designs for weaknesses and has brought out a new design in which those weaknesses have been corrected, he has to combat the inertia of buyers, and their disinclination to think, in order to present the merits of his new machine. Without a real engineering foundation for a decision one hopes that the buyer is seriously considering economic aspects. However if the buyer has no real foundation for forming a judgment, a salesman cannot expect to sell a machine on the economic basis of profit for the purchaser.

From reports that reach my offices, the theory in the automobile industry seems to be not to purchase a machine unless it pays over 100 per cent income. That is a high rate of income, a higher rate than is earned by any other capital invested in the plant. Some buyers are apparently foolish enough to deprive their companies of 25-per cent a year investments because of this 100-per cent obsession.

If you have a reasonable engineering basis for your judgment in purchasing a machine-tool, you will be able to gage or score the machine. Then when the financial executive says to the production executive, "The price is too much," the production man can say, "You have the facts in black and white." Facts should be the dominating factor in judging investments. The right type of production executive is able to show why he recommends a machine as an economic investment and that the production and maintenance costs justify his perference for one machine over some other.

Would Rate Machines Like Fire Risks

One of my hopes is that some day some engineers will devise a machine-scoring system like the fire-insurance rating-system. The underwriters have standards whereby they rate a building as an insurance risk. They add a certain number of points to the standard rate for each deficiency, and the premium increases according to the rating. They also make deductions for improvements. Some day perhaps, the machine-tool builders and users will see the wisdom of doing that with machine-tools. All sensible buyers do some rating now, consciously or unconsciously, based on impressions gained in some way. Why not systematize the recording of these impressions? Better still, when facts are collected, as Mr. Kelso has gathered them, the facts themselves provide a basis that is much better than any haphazard method of forming judgments.

To do this, it is necessary that the production man call on someone in the plant who has sufficient statistical sense to gather the right kind of information and to correlate it. That is a very practical sort of statistical work. A field exists for good statistical methods in the study of maintenance and relative costs in machine-tool operation.

Users Can Help With Statistics

Forrest E. Cardullo[14]:This paper is a first-class example of the kind of cooperation the machine-tool builder needs to make himself of more use to the automotive engineer. In a way, a customer's plant must be our laboratory. No great principles of physics or electricity or chemistry trouble us. The kind of research work we need is exactly the kind that Mr. Kelso is doing. We are not in position to do that work as well as our customers can do it. If we can induce a number of customers to do that sort of work and to lay the figures before us, we shall be able to design better machine-tools. For example we could learn considerable from the failures of vertical drilling-machines if we could have a systematic analysis of the troubles due to different types of construction, and the information would be of great advantage both to the builder and the user of the machine.

The only fault I can find with Mr. Kelso's paper is that the author does not go far enough in pointing out the causes of the failures but analyzes them only in a general way. To be of the greatest use the analysis should be carried farther and show what parts had to be repaired and why they failed.

The Problem of Rebuilding Machines

The high cost of rebuilding a machine appears to be a serious problem. When the cost runs up to half the price of a new machine, it is not economical to rebuild it. If it requires rebuilding at such a cost after a period of 6 or 8 months or 1 year, it would seem that something is radically wrong, and that either the machine needs redesigning or the method of use or system of maintenance is faulty. Perhaps if the builder were called in he would be able to point out ways in which the machine could be handled so that it would not have such a high rate of depreciation.

A Member:—I think this is a line along which many of us should be doing considerably more than we are. One thing I do not exactly understand is what Mr. Kelso means by "frequency percentage."

Mr. DuBrul:—He has a footnote reading,

> Frequency is defined as the average percentage of the number of machines in use that are taken down for any repair during the month.

If you have 100 machines and in the course of a month you have to make some repairs on 8 of the machines, the frequency percentage is 8.

W. G. Careins[15]:—We have found in the Nash and Ajax plants that probably the best plan is to return the machines to the builders from time to time and have them rebuilt. A considerable number of vertical drilling and other machines have been sent back. The builders could rebuild them and put them into first-class condition better than we could have done it. I should like to hear from others regarding this and as to the attitude of the machine-tool builders.

Furnishing Repair Parts a Nuisance

Mr. DuBrul:—We have had considerable discussion in our Association meetings on the question of repairing and rebuilding. Many machine-tool-shop managers have been deceiving themselves with the idea that theirs could be a production industry like the automobile industry. If they attempt to organize their business along that line they will overproduce their market. The furnishing of repair parts for old machines now becomes a nuisance. They do not make any money at it. When a repair part is ordered, it takes the time of real engineers to make sure that the shop gets out the right pattern and casting. Then the shop has to carry it through as a special order. That runs the cost up tremendously. In our industry 100 machines of a kind is a large lot. That sounds ridiculous to you who deal in large quantities every day, but it is a fact. Slipping

[14] Chief engineer, G. A. Gray Co., Cincinnati.
[15] M.S.A.E.—Production engineer, Ajax Motors Co., Racine, Wis.

those odd repair parts into a production schedule of 100 machines becomes very costly. The machine-tool builder does not have the courage to even figure all of these costs, much less to charge for them.

Rebuilding Does Not Pay Builders

The question of rebuilding arises frequently. Some companies will rebuild machines of their make when times are dull. This work is taken to provide their men with employment, but when they are busy they need all their men on new work and will not rebuild machines. It is a matter of organization in the individual machine-tool shop. We have no reliable information as to whether or not to do rebuilding is good practice.

Rebuilding sometimes creates a bad commercial condition. A firm takes a machine in trade, rebuilds it and sells it to some other customer. The lying buyer creates a bad spirit of business unfairness by telling another machine-tool company, "Your competitor gave me a new machine at a second-hand price." It is not ethical and does no good to either party to the transaction. To demoralize the ethical standard of any industry is not good business.

I do not know of any company at present that will rebuild any of its machines. When skilled labor is scarce. as it is now, and the demand is in excess of the supply the builders cannot secure enough machinists of the kind needed. Can you then expect those companies to rebuild their machines at any price? Not at all. They cannot turn out the volume of new production demanded right now by the automotive industry.

Fluctuations in Demand Are Excessive

If large machine-tool users were to forecast their requirements better than they now do, they would serve their own interests much better than they are now serving them. It is both a commercial and a manufacturing waste, for which users are responsible and for which they must pay. There are dull times when the hunting for new orders for machine-tools is very poor. The user waits until he is confronted by the immediate need for new machines before he places orders for them. Statistical studies that we have been able to make show that the fluctuation in activity in the machine-tool business is more violent than that in iron and steel. The demand for machine-tools at any time is constituted of two elements. The first is represented by orders for replacing the old equipment that has become obsolete, not so much because it has worn out as because some newer and better machines that have been brought out make the old types uneconomical. The second element is the expansion demand resulting from the building and equipping of new shops, or the starting of entirely new industries, like the recently created radio business. When the automobile business was young, it also furnished an expansion demand. The managers of a rapidly expanding industry cannot forecast their sudden demand. So when they want certain machines the builders are not always able to supply the ones they want and they are obliged to take something else. When the demand slackens, they do not dispose of those machines and replace them with the kind they prefer. The automobile trade waited until the automobile business recovered and then demanded new machines all at once.

The machine-tool industry has not been able to train enough good machinists to supply all the tools now needed by the automotive industry. In 1914, which was a year of depression, it did not put on any apprentices. Then came the war and post-war boom and it could use handymen on the relatively large production of given types of machine, so it did not train machinists then. In the depression of the last few years the automobile builders picked up all of the floating supply of machinists. The secretary of a metal-trade association told me that several automotive companies swept-up all of the floating supply of Cincinnati's machinists during the depression. Now they want more tools from the machine-tool industry and they wonder why the machine-tool builders cannot turn them out. The reason is that the supply of skilled machinists in this Country is less than the demand.

Machines Must Pay for Themselves Quickly

Erik Oberg[16]:—I used to think that it was unreasonable for the automobile companies to require the machines that they buy to pay for themselves in so short a time, until I had an opportunity to investigate a few months ago and found that the peculiar conditions of the automobile industry would make it very uncertain to buy machine-tools on a basis of their paying for themselves in from 3 to 6 years. As an example of what happens, a special type of milling-machine was bought for $9,800. The machine was run for 1 week and then the design of the automobile engine was changed and the machine was entirely useless for the new model. That is the sort of thing confronting the automobile builder and I can understand that when he buys machine-tools he wants to protect himself by making sure that the machine will pay for itself in a very short time and will not become useless for his work. The automobile builder must get his money out of the machines very quickly, particularly from semi-special and special machines, because something may happen to make them useless. In one instance an equipment that cost over $200,000 was used an average of only 6 months and then sold for a little more than $30,000.

Chairman John Younger[17]:—I think Mr. Oberg has hit the nail on the head. Machines are selected for the production of certain models. Mr. Litle, who has been nominated as President of the Society for next year, is one of the leaders in a movement to abolish the yearly-model practice and is doing much toward helping the industry and its suppliers.

Plants Need Machine-Tool Inspectors

Eugene Bouton[18]:—I was interested in the fact that Mr. Kelso had tabulated statistics indicating how often machine-tools were repaired. The automotive industry has inspectors for quality of production, safety inspectors and so on, and it would seem that an inspector for machine-tools and equipment would be proper, and considerable saving shown over a given period in the maintenance and repairs of machine-tools. It seems to me that in a large shop having from 500 to 600 machines it would pay to have a man, who thoroughly understands machine-tools, make frequent inspections to determine what repairs and replacements are necessary to keep the machines going, thereby preventing a complete breakdown later.

My own opinion is that it does not pay to rebuild a machine-tool when the rebuilding costs half the price of a new machine. It has been the practice of many machine-tool builders to quote on that basis, and when the freight and handling charges and the reinstallation cost

[16] M.S.A.E.—Editor *Machinery*, New York City.

[17] M.S.A.E.—Publisher, *Automotive Abstracts*, Cleveland.

[18] M.S.A.E.—Supervisor of time study, Chandler Motor Car Co., Cleveland.

of the machine after the repairs have been made have been added thereto, it would be cheaper in the long run to purchase a new machine. If the trade-in allowance on a new machine were only one-fourth of its original value or if the machine could be sold in the second-hand market for such an amount, a substantial saving at the end of a 5-year period over the rebuilt machine would be shown. The above, of course, does not apply to heavier types of machine that run upward of $15,000 or $20,000, but applies to machine-tools under $10,000. I am not stating these as facts but as my opinion.

CHAIRMAN YOUNGER:—One shop that I was in made a practice of allowing a certain percentage of its machines to be constantly out of service and when the master mechanic went around he would pick out one of these machines and overhaul it. I know of other shops that have a policy of regularly reconditioning machines on the old theory that "a stitch in time saves nine."

INSPECTION METHODS

BY C. J. ROSS[19]

ABSTRACT

WITH the passing of the apprenticeship system and the introduction of the present method of employing unskilled labor on a piecework basis for assembling, careful inspection has become a necessity. Under these conditions, the only way in which the product can be held to the required standards is to make the component parts fit accurately. If the inspection is adequate, parts can be held to closer limits and cheaper labor can be used in the assembling process.

Believing that no reason can exist for failure to maintain standards of accuracy if the ratio of the number of men engaged in production to one inspector does not exceed 15 to 1, the officials of the Buick Company have worked out a system, similar in many respects to a budget, in which a certain ratio of production hours to inspection hours is allowed in each plant, the number depending upon the nature of the work and varying from about 10 to 1 in the engine plant to about 34 to 1 in the gray-iron foundry. In actual operation these ratios were usually reduced.

The inspection department is divided into four divisions, each of which is under the jurisdiction of a divisional inspector who reports directly to the general superintendent. All major parts are given 100-per cent inspection. Parts purchased from other companies are given 10-per cent inspection, but if any are found faulty all are checked.

The process of inspection is followed from the time the incoming raw material is checked for quality by the metallurgical department, as it passes through the various steps of production, until it reaches the final assembling line. In general, inspection by pin and snap gages of the go and no-go type has been found most satisfactory, because this method gives quicker inspection and establishes definite limits beyond which parts are not allowed to pass.

Many of the testing devices employed are similar to those in common use in the industry but others have been specially devised to meet particular needs. Several of these containing noteworthy improvements are described in detail and illustrated. Extensive application is made of spark testing, both of the raw material and of parts coming from the heat-treatment. By this method not only can steels that may have become mixed be readily sorted, but the carbon-content and the presence of inclusions can be quickly and accurately determined, and the percentages of nickel, chromium, tungsten and vanadium present in an alloy-steel can be fairly closely estimated. Among the special devices described are those for inspecting crankshafts, cylinder bores, flywheels, camshafts, differential carriers, steering-knuckles, universal-joint yokes, chassis springs, the inside diameter of steel flywheel rings, wheel felloes, and the toe-in and camber of axles.

THE DISCUSSION

PAUL M. MUELLER[20]:—What methods are used to maintain a fixed quality standard that cannot very well be measured, such, for instance, as the quality of noise, in examining assembled transmission-units?

Whenever work is coming through in good shape and no trouble is being experienced, the standards of quality creep up until after a period, material that would have been considered excellent at the beginning of the period is no longer considered satisfactory. At least, that has been our experience on the few cases in which we have had contact with things of this sort.

R. B. SCHENCK[21]:—To my knowledge, nothing in the way of mechanical noise-indicators has ever been worked out, but attempts have been made. The amount of transmission-gear and timing-gear noise that occurs is simply a question of the personal opinion of the expert who is engaged in that work. A man with a good ear who has listened to thousands of transmissions and timing-gears is able to set a certain standard. When a number of inspectors are engaged in that kind of work, they work close to one another and so have a standard that depends on personal opinion which is fairly well established.

MR. MUELLER:—My contention is that a man under those circumstances cannot set a standard. It is not an absolutely fixed standard but it is the best that can be done. Then is there no method of holding them to it?

MR. SCHENCK:—There is no definite standard.

CHAIRMAN A. H. FRAUENTHAL[22]:—I think that is a salient point. When the work is coming good, the standards grow higher and there seems to be no way of stopping this.

DR. ANCEL ST. JOHN[23]:—Would not a device for establishing a standard for a certain noise be of considerable value and would it not warrant a certain amount of investigation and development work?

MR. SCHENCK:—Anything that would determine the amount of noise accurately, cheaply and quickly and eliminate the human element would be a big step in the right direction.

CHAPIN HOSKINS[24]:—Some work of this kind has been done at Ann Arbor, Mich., and I believe an automatic recorder is being developed at the River Bank laboratories of Geneva, Ill.

CHAIRMAN FRAUENTHAL:—In making any kind of instrument to check transmission noise, the problem would be fairly simple if it were merely a matter of registering the quantity of noise; but in an automobile a peculiar situation exists. Noises of different magnitudes are not necessarily objectionable in the ratio of

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[21] M.S.A.E.—Metallurgical engineer, Buick Motor Co., Flint, Mich.
[22] Jun.M.A.E.—Chief inspector, Chandler Motor Car Co., Cleveland.
[23] Consulting physicist, New York City.
[24] Managing editor, *Factory*, Chicago.

their magnitude but rather according to the type of the sound. For instance, a rough deep sound of a grating nature gets on one's nerves, whereas one of higher pitch near middle A and smoother is not so objectionable.

MR. SCHENCK:—The question of resonance enters, also the size and shape of the body, the crankcase and the transmission case, the material of which they are composed, the weights and many other factors of that sort.

DR. ST. JOHN:—That is quite true in a finished car. The problem that we are considering is the individual contributions from the different elements in the car. These, it seems to me, could be valued separately by some sort of mechanism that would tell not only quantity, but the quality that is allowable.

MR. MUELLER:—I might suggest what has already been done under the auspices of the American Medical Association. In stethoscopes, they use amplifying devices with a sound-recorder, such as a telephone transmitter. The sounds are sent through radio amplifying tubes and band and line filters, and the like, are inserted that suppress different components of the sound, so that the different types and degrees of noise in the human heart are segregated. The same methods might be applicable to transmission work.

R. H. SHERRY[25]:—Standardizing noise with reference to quality will be done with difficulty until it is possible invariably to tell the location of the noise. Some years ago production of a high priced car was held up on account of noise that could not be traced to a definite source. Sound-proof rooms were built to see if the cause could be detected but without success until someone accidentally placed his hand on the transmission case which stopped the sound. On lifting the hand the noise was again heard. The transmission case had been changed slightly, the effect being that of a sounding board. Resonance is often more responsible for audible sounds than the original source from which the sound waves start.

M. H. LEAGUE[26]:—Do you inspect the Delco system 100 per cent?

MR. SCHENCK:—Yes.

H. F. BLANCHARD[27]:—An instrument[28] used by the Timken Roller Bearing Co. for measuring the volume of sound in bearings was described by F. A. Firestone at the last Annual Meeting.

R. I. MOWRY[29]:—Referring to the percentage chart given in the paper, I notice that in the axle department the ratio of the number of production hours to inspection hours during 1 month went up to 15 and the next month went down to 12. What is the cause of this increase in the number of inspection hours, assuming that new models are not being brought out?

MR. SCHENCK:—Possibly a little fluctuation in the production.

MR. MOWRY:—The reason that I bring up that point is that under ordinary circumstances, when a method has been laid out and the full number of men is working, a certain number of inspectors is ample, but if an epidemic of careless work suddenly appears, we in the inspection department know that the production foreman is not checking the men.

As a rule, if an investigation is made, it will be found that the quality of the work passed by the inspection department has not changed. Sometimes it does change but in most cases it does not. Poor work is generally due to the fact that the men on the assembling lines are not held in check by the production foreman and poor work is allowed to get by them.

I am wondering whether any rigid rules have been laid down by which production foremen are compelled to check their assemblers; and how often they inspect and follow up the work themselves.

CHAIRMAN FRAUENTHAL:—We have not found that it can be absolutely controlled. A few years ago we introduced into our shop a group piece-work system that looked very discouraging from this standpoint at the time. We pay a certain amount of money to a fairly large group of men for each completed assembly or machine job. On the chassis line, for instance, we have about 100 men and pay so much for each completely assembled chassis coming from the end of the line. We do not furnish very much detail inspection to that line. Suppose that at the end of the line the final inspector finds a few defects that necessitate taking the chassis apart and fixing it. The man who did the bad work has the other fellows on his back because they are forced to dismantle their good work to get at his bad work. We find that if we make the men take a few cars apart once a month or so, the men in the line virtually become inspectors and will not allow the men ahead of them to do poor work. This method of discipline means a loss to their pockets and is the best way we know to control the situation.

QUESTION:—What are the limits of tolerance on the cylinder-bores? As I understand it, you allow 0.0005-in. taper and 0.0005-in. out-of-roundness.

MR. SCHENCK:—Yes.

QUESTION:—Are the pistons fitted by size or by the code number?

MR. SCHENCK:—The pistons are marked with a code number for classification and are fitted according to size.

QUESTION:—What is the piston clearance?

MR. LEAGUE:—About 0.003 in.

MR. MUELLER:—Is any device in use for gaging a cylinder-bore by integrating its area, for instance, a pin-type gage with a number of pin pistons pressed out radially against the clinder-wall from a common reservoir or cylinder of mercury. Has any attempt been made to get the effective area of the cylinder-bore rather than the unit dimensions of it? Would there be any advantage in doing so?

MR. SCHENCK:—I do not think so. The micro-gage and the paddle gages that we use are very accurate. With a paddle gage inspectors can feel a difference of 0.0001 in.; a difference of 0.0005 in. is very noticeable.

CHAIRMAN FRAUENTHAL:—You mentioned in the early part of the paper that the inspection is divided into four divisions and that the four divisional inspectors report to the general superintendent but that you have no chief inspector.

MR. SCHENCK:—There is no chief inspector. The four divisional inspectors report directly to Mr. Ross, the general superintendent. The metallurgical inspectors report to the chief of the metallurgical department; but that is a separate department.

MR. LEAGUE:—In feeling tenths of thousandths in testing cylinders, the expansion and contraction at different temperatures must affect the cylinders and the gages.

CHAIRMAN FRAUENTHAL:—The temperature would not

[25] M.S.A.E.—Consulting metallurgical and industrial engineer, Evanston, Ill.

[26] Trouble department, Electric Autolite Co., Toledo.

[27] Technical Editor *Motor*, New York City.

[28] See THE JOURNAL, February, 1925, p. 121.

[29] Chief inspector of sheet metal stampings, Studebaker Corporation of America, South Bend, Ind.

make much difference because the gages and cylinders have about the same temperature.

MR. LEAGUE:—While it is not commercial to talk in tenths of thousandths, I still maintain that to measure in tenths of a thousandth 68 deg. fahr. must be the temperature.

MR. SCHENCK:—An important point in the use of gages is to have both the gage and the work at the correct temperature.

MR. HOSKINS:—Some industries have been trying to pay inspectors on the incentive basis and some have found that they can do so without affecting the quality. Has Mr. Scheneck ever tried paying inspectors on an incentive basis?

MR. SCHENCK:—I do not think that it can be done. Inspectors formerly were paid on a piece-work basis, but that proved unsatisfactory. The most satisfactory way is to have a man work at the regular day-rate. In the incentive system you mention, I assume that a man would be paid according to the number of defective parts.

MR. HOSKINS:—A half-dozen or more companies have tried that and other methods. The Holeproof Hosiery Co. has a pyramid of inspectors. One or two inspectors at the top check a certain percentage of the work of the inspectors on down the line. By checking them in that way the company has found it safe to pay for greater speed.

MR. SCHENCK:—To the best of my knowledge that is not being done in the automobile industry.

SHEET STEEL FABRICATION

BY SYD SMITH[30]

ABSTRACT

ECONOMY is the watchword of the automotive industry. Under the present conditions of keen competition, the race has become a question of the "survival of the fittest." Inasmuch as the general appearance of an automobile depends mainly upon the sheet or strip steel used in its construction, the weight of which comprises from 25 to 30 per cent of the total weight of the car, and as approximately 65 per cent of the cost of a piece is in the stock, the opportunity to reduce costs by conserving the stock is obvious.

Keeping this point in mind, particular attention has been devoted to the utilization of the scrap from large pieces in the manufacture of small parts. Scale drawings are made of each size of sheet steel received at the factory and a card-index quickly gives information as to the possibility of making any new piece from scrap material. With a view to conserving every part of the material, several small pieces are welded together into such regular sized and shaped sheets as would be suitable for larger parts such as dust-pans, seat-pans, regulator-boards, and the like.

So great has been the progress within the last few years in the production of parts from sheet steel that they have largely superseded castings and forgings, particularly those used as body parts. But greater possibilities, however, still exist in this direction, even to the extent of making phaeton tops from a single sheet of welded steel.

The average cost of the finished sheet-metal parts of an automobile, including the enameling of the parts, is said to be about $0.075 per lb. This is about the same as that of gray-iron castings, about one-half that of malleable castings, about 30 per cent less than that of forgings, and only about one-tenth that of aluminum castings. Although sheet-metal parts are on a par with iron castings when compared on a basis of the cost per pound, they make a still better showing when compared on a basis of strength cost.

To cite a concrete example and show a method of producing sheet-metal parts, the progress of several automobile parts, including the shroud, tonneau and side-rails, is followed during their fabrication, and the manner of assembling the side-rails and some minor parts into the chassis frame is described.

[30] Superintendent of stamping, Studebaker Corporation of America, South Bend, Ind.

[31] Engineer, International Motor Co., Allentown, Pa.

[32] Chief inspector of sheet metal stampings, Studebaker Corporation of America, South Bend, Ind.

[33] A.S.A.E.—Assistant to the vice-president, American Sheet & Tin Plate Co., Pittsburgh, Pa.

[34] M.S.A.E.—Plant superintendent, Hudson Motor Car Co., Detroit, Mich.

THE DISCUSSION

W. P. MICHELL[31]:—On the question of steel sheets, Mr. Mowry, do you make a practice, on your body panels, of buying sheets cut to exact sizes, or do you cut the panels out of stock sheets?

R. I. MOWRY[32]:—We buy them as near the size as we can.

MR. MICHELL:—In every case?

MR. MOWRY:—In every case; that is, we buy them so that we can get one or more units out of a single sheet and have the least possible amount of scrap left. Sometimes sheets are bought with a view to cutting various sized blanks. We might get two of one kind, and one of another.

W. A. IRVIN[33]:—Before you installed roller-levelers, did you have any trouble with stretcher strains in the stampings?

MR. MOWRY:—We did, occasionally.

MR. IRVIN:—Do you find that, if you roller-level the sheets, say 3 or 4 days or a week before you are ready to stamp them, you have more trouble stamping them than immediately after roller-leveling them?

MR. MOWRY:—Continuing the process immediately after roller-leveling is the better practice. In that connection I might say that we find some steel that must be passed through two or three times, if the steel is bad.

R. Z. HOPKINS[34]:—Do you have any trouble with the welding? I understand that the welds are butt welds.

MR. MOWRY:—They are butt welds, flash welds. The metal is brought together with the current applied. The current is shut off automatically at the end of the stroke.

MR. HOPKINS:—Are the parts finished before they are put on the body, or afterward?

MR. MOWRY:—They are finished before they are put on the body.

MR. HOPKINS:—How much hand-finishing do you do on the body after it has been assembled to the frame?

MR. MOWRY:—That depends on the care that is taken in handling the panels and applying them to the frames. In our practice, all body-panels are made as nearly perfect as we can make them before they leave the fabricating department. The panels are assembled to the frame in another department. All panels, when they leave our department are finished to take paint or enamel.

MR. HOPKINS:—Then the only finishing that is done on the line is the removal of dents or scratches that have

occurred in the process of assembling the body panels?

MR. MOWRY:—Yes.

MR. HOPKINS:—Do any of the welds fail?

MR. MOWRY:—No, we had a little trouble at first with some of the experimental work cracking and breaking, but this was corrected before regular work was started.

MR. HOPKINS:—Under vibration?

MR. MOWRY:—Yes.

MR. HOPKINS:—Have you made any vibration tests on the welds?

MR. MOWRY:—Vibration tests have been made in the laboratory by a twisting machine. The body is simply bumped, rocked and twisted for many hours in an endeavor to break it; not only these welds, but other joints are constantly tested in that way.

MR. MICHELL:—Mr. Mowry made the assertion that on certain body panels dies were used. What is done in that regard?

MR. MOWRY:—Because of the variety of current models, we found it was not practicable to use large dies on small runs of certain kinds. Because of the many body styles that we build, it would hardly pay to do so. Whether blanking, roller shearing or sawing out the blanks is the most economical method of doing the work is a question that the head of the department must decide.

W. G. CAREINS[35]:—Have you ever given thought, when blanking side-rails, to the use of just enough material to make the side-rail on edge, and upsetting it, to get the kick-outs? Some companies follow that practice.

MR. MOWRY:—No.

MR. CAREINS:—What sort or cooperation do you get from the engineering department in designing stampings to use up scrap?

MR. MOWRY:—The heads of our manufacturing departments are called in to consult with the engineers, when new parts are designed or new models brought out, with a view toward economical manufacturing.

PRODUCTS AND BY-PRODUCTS OF FOREMEN'S CONFERENCES

BY FRANKLIN T. JONES[36]

ABSTRACT

THE technique of introducing, conducting and auditing foreman training is presented in detail, reference being made to four main sources of information from which liberal quotations are made and commented upon. Actual methods used and results from foreman training are related by the author, who also discusses the fine points of successful procedure.

Among the important features of educating foremen so that they can attain higher levels of average practice are the selection of competent conference leaders, proper comprehension of the foreman's viewpoint and the inspiration of his confidence in the sincerity of the management. Conference topics are suggested and it is advised that these be presented to successive small groups of men who constitute a fair "sample" of the organization rather than to all the foremen simultaneously.

Charts illustrating how the higher levels of intelligence are attained by such means are presented and methods of follow-up that assure permanence of the training, with statements of proper subjects for study to supplement the conferences, are explained.

Means for deducing tangible results from the foregoing educational process are given in detail. Direct results are considered to be products, and indirect results are counted as by-products; from them, the men can be rated and ranked. A summary is made in conclusion.

THE DISCUSSION

R. D. BUNDY[37]:—The discussion seems to center on whether an instructor should be brought in from the outside or whether he should be drawn from the executive force, to take charge of this work of foreman training. A few years ago when I was assistant manager of the Cleveland Mazda Lamp Works, a representative of the Board of Education called on us relative to putting on a series of foremen's conferences in our plant. We thought well of the idea but were cautious as to whom we would allow to come in and talk to our foremen. After an interview with the proposed conference leader, it was decided to start the conferences. I became very much interested in this work because I saw the advantages of having someone outside of the management guide these discussions. He was able to put across ideas in an impartial manner that had weight with our foremen. We felt that in so doing he was driving home principles which the management had been advocating.

The men became more receptive and open-minded because they would discuss these problems among themselves and would then decide among themselves whether or not the things taught were right. After their own discussions and conclusions they would not think the principles impractical.

It was our practice to hold conferences regularly at which factory problems were discussed before the outside man started his conferences. These conferences were also continued during and after he completed his course.

Later, I had the opportunity of taking up this work in Cleveland. I have had conferences with leaders in Mr. Jones' and in several other plants and so have observed training methods from both sides. I believe there is a distinct advantage in having a conference leader come from the outside to hold foremen's conferences such as was advocated in Mr. Jones' paper. This man should be one who is able to do the work properly. He should not go into a conference with the thought in mind of telling men how to run their departments but should emphasize the fact that the conferences are to be an exchange of ideas which will be of benefit to all. The value of these conferences will, therefore, depend upon how the discussions are led and to what extent the foremen enter into them.

[35] M.S.A.E.—Production engineer, Ajax Motors Co., Racine, Wis.
[36] Supervisor of training, White Motor Co., Cleveland.
[37] Supervisor of foreman training, Board of Education, Cleveland.

Applicants *for* Membership

The applications for membership received between Jan. 15 and Feb. 15, 1926, are given below. The members of the Society are urged to send any pertinent information with regard to those listed which the Council should have for consideration prior to their election. It is requested that such communications from members be sent promptly.

AFF, LOUIS WILLIAM, owner, L. W. Aff Cylinder & Machine Works, *San Francisco.*

AMBROSE, ROY B., associate professor and head of automotive department, Carnegie Institute of Technology, *Pittsburgh.*

BABIKOFF, VLADIMIR VASILIEVICH, sales engineer, International Motor Co., *Long Island City, N. Y.*

BACHMANN, JOSEPH E., motor-truck fleet owner, Coolidge Point, *Magnolia, Mass.*

BALDWIN, HOWARD D., engineer, Connecticut State Highway Commission, *Hartford, Conn.*

BECHTEL, G. E., production engineer, White Motor Co., *Cleveland.*

BENNETT, EDWARD C., lubricating engineer, Empire Refineries, Inc., *Chicago.*

BINDER, JOHN A., engineer, Dodge Bros., Inc., *Detroit.*

BOYD, PAUL M., production engineer, Curtiss Aeroplane & Motor Corporation, *Garden City, N. Y.*

BROWN, RISDON D., experimental garage foreman, Oakland Motor Car Co., *Pontiac, Mich.*

BUETHE, AUGUST, JR., superintendent, Lexington Machine Co., *Rochester, N. Y.*

BUTLER, ROBERT W., engineer, Robert W. A. Brewer, *Ridley Park, Pa.*

CHANDLER, M. G., chief engineer, Vacturi Carburetor Co., *Chicago.*

CONDAK, Y., industrial and mechanical engineer, White Motor Co., *Cleveland.*

COOKE, HARTE, engineer, McIntosh & Seymour Corporation, *Auburn, N. Y.*

DE ROZA, ROBERT J., assistant manager, Federal Motors Co. of California, *San Francisco.*

DICKEY, RICHARD J., chief engineer, Gabriel Snubber Mfg. Co., *Cleveland.*

DIEDERICHS, JOE B., draftsman, Diamond Motor Parts Co., *St. Cloud, Minn.*

DOMAN, CARL TEMPLAR, engineer, H. H. Franklin Mfg. Co., *Syracuse, N. Y.*

EDSON, FRANKLIN C., president, Asbestos Mfg. Co., *New York City.*

EMERSON, JOHN R., testing laboratory engineer, Continental Motors Corporation, *Detroit.*

EVANS, R. J., vice-president, Asbestos Mfg. Co., *Huntington, Ind.*

FORE, HARRY A., engineer and salesman, Philadelphia Storage Battery Co., *San Francisco.*

FRANCHIMONT, HENRI A., aeronautical engineer, Atlantic Aircraft Corporation, *Hasbrouck Heights, N. J.*

FUNNELL, HARRY M., engineer, Spicer Mfg. Corporation, *South Plainfield, N. J.*

GALBRAITH, L. E., engineer, Toledo Steel Products Co., *Toledo.*

GARFIELD, JAMES ABRAM, vice-president, Christensen Air Brake Co., *Cleveland.*

GENUNG, GILBERT, transportation supervisor, Public Service Co. of Northern Illinois, *Chicago.*

GIROUX, GEORGE A., inspector of naval aircraft, c/o Loening Aeronautical Engineering Corporation, *New York City.*

GLOWNEY, JOHN F., draftsman, Hupp Motor Car Corporation, *Detroit.*

GOLDSMITH, C. S., assistant superintendent of transportation, Brooklyn Union Gas Co., *Brooklyn, N. Y.*

GRIFFIN, JAMES C., field service representative, Ruckstell Sales & Mfg. Co., *New York City.*

HANCOCK, JAMES EDWIN, chief engineer, Morgan & Wright, *Detroit.*

HANDS, RONALD C., assistant superintendent, Eastman Kodak Co., *Rochester, N. Y.*

HARRINGTON, N. T., secretary, treasurer and general manager, Simplex Piston Ring Co., *Cleveland.*

HASELTINE, C. W., secretary and treasurer, Mack Trucks, Inc., *New York City.*

HEIMBERGER, LOUIS J., draftsman, Hupp Motor Car Corporation, *Detroit.*

HELFRICH, HENRY J., truck experimental engineer, Pierce-Arrow Motor Car Co., *Buffalo.*

HERLIHY, FRED W., engineer, Vacuum Oil Co., *Boston.*

HINSHAW, JOHN C. W., president and treasurer, Airbuoy Co. of Illinois, *Chicago.*

HIRSCH, ROBERT R., district sales manager, S. K. F. Industries, Inc., *Buffalo.*

HOLLANDSWORTH, G. C., chief engineer, Reliable Trucks, Inc., *Chicago.*

HUMISTON, H. A., electrical engineer, National Carbon Co., Inc., *Cleveland.*

IINUMA, TOMOSHIGE, First Tank Battalion, *Kurume-Shigai, Fukuoka-Ken, Japan.*

JARRELL, JOHN E., service engineer, Ross Gear & Tool Co., *Lafayette, Ind.*

JARVIS, HAROLD, vice-president, Transportation Guarantee Co., *San Francisco.*

JOHANSEN, HARRY V., sales engineer, Brandenburg Bros. & Eccleston, *Detroit.*

JOHNSON, ANDREW B., maintenance superintendent, Peoples Motor Coach Co., *Detroit.*

JOHNSTON, CHARLES F., partner, Super Auto Products Co., *San Francisco.*

JONES, CLYDE RANDOLPH, owner, Service Garage, *Pasadena, Cal.*

JUST, JOHN, toolroom foreman, Excelsior Motor Mfg. Co., *Chicago.*

KING, HENRY T., draftsman, Hupp Motor Car Corporation, *Detroit.*

KLINE, HARRY D., advertising manager, Continental Motors Corporation, *Detroit.*

KNOX, ELMER G., factory manager, Yellow Coach & Truck Mfg. Co., *Chicago.*

LEIGHTON, HENRY D., service man, Myron D. Kidder, Inc., *Portland, Me.*

MCARTHUR, A. S., superintendent of garage department, Toronto Transportation Commission, *Toronto, Ont., Canada.*

MACCRACKEN, WILLIAM P., JR., lawyer, Montgomery, Hart & Smith, *Chicago.*

MADDEN, EDWARD J., engineer, Madden Compensated Brake Co., *Detroit.*

MARBLE, H. C., assistant general service manager, White Co., *Cleveland.*

MATTINGLY, L. H., president, Blood Bros. Machine Co., *Allegan, Mich.*

MICHELL, ROY G., secretary, treasurer and general manager, Eclipse Interchangeable Counterbore Co., *Chicago.*

MILLER, FLOYD D., purchasing agent, Bryan Steam Corporation, *Peru, Ind.*

MITCHELL, RALPH, draftsman, Hupp Motor Car Corporation, *Detroit.*

MOHNS, LESLIE, draftsman, Rockford Drilling Machine Co., *Rockford, Ill.*

MOORMAN, A. H., vice-president and treasurer, Wills Sainte Claire, Inc., *Marysville, Mich.*

MUNCEY, JOSEPH J., model maker and assistant engineer, Curtiss Aeroplane & Motor Corporation, *Garden City, N. Y.*

NEELANS, HAROLD G., experimental engineer, Dodge Bros., Inc., *Detroit.*

NEWELL, GEORGE A., JR., branch manager, Fruehauf Trailer Co., *Detroit.*

O'LAUGHLIN, F. M., Eastern sales manager, Climax Engineering Co., *New York City.*

PACKARD, JOSEPH A., engineering secretary, White Motor Co., *Cleveland.*

PHILLIPS, HAROLD IRVING, sales engineer, Flintlock Corporation, *Detroit.*

RAYNIAK, JOSEPH G., works manager, Johnson Motor Co., *South Bend, Ind.*

REES, GEORGE G., partner, Super Auto Products Co., *San Francisco.*

SCHENCK, PAUL F., instructor, Board of Education, *Dayton, Ohio.*

SCHLARMANN, THEO H., technical salesman, Pressure Proof Piston Ring Co., *Boston.*

SCHMIDT, HERMAN H. A., JR., assistant superintendent of Central Division, Public Service Transportation Co., *Elizabeth, N. J.*

SCOTT, EVERET CHAPIN, Secretary, Sterrett & Co., *City of Washington.*

SHOEMAKER, L. F., assistant Eastern sales manager, Buda Co., *New York City.*

SIMMONS, O. G., general manager, National Tool Co., *Cleveland.*

SKINNER, CLARENCE O., engineering, Motor Wheel Corporation, *Lansing, Mich.*

SMALL, WAYLAND B., master mechanic, Lang Transportation Co., *Los Angeles.*

SMITH, A. T., sales manager, Mack Truck Co., *San Francisco.*

SMITH, M. A., president, McKee Glass Co., *Jeannette, Pa.*

SOROUR, M. H., mechanical transport engineer, Ministry of Communications, *Cairo, Egypt.*

STARLING, F. M., lubricating and fuel salesman, Standard Oil Co. of New York, *New York City.*

STAY, T. D., manager and technical supervisor, U. S. Aluminum Co., *Cleveland.*

STEIN, FRED WILLIAM, tool supervisor, Fairbanks, Morse & Co., *Beloit, Wis.*

SUSSIN, VICTOR, engineer, Stewart-Warner Speedometer Corporation, *Chicago.*

SUTTKUS, HENRY E., foreman of engine department, Elco Boat Works, *Bayonne, N. J.*

THOMSON, NORMAN A., engineer, Hayes Wheel Co. of Canada, Ltd., *Chatham, Ont., Canada.*

THORNSTON, G. A., mechanical engineer, Grindle Fuel Equipment Co., *Harvey, Ill.*

THORPE, CHARLES, service superintendent, Mack Truck Corporation, *San Francisco.*

TIMOTHY, HUGH W., managing engineer, Vacuum Oil Co., *Boston.*

TWINING, EARL S., assistant to chief engineer, Champion Spark Plug Co., *Toledo.*

VAN GUILDER, WALTER, assistant chief engineer, Stewart-Warner Speedometer Corporation, *Chicago.*

VISCHER, ALFRED, constructive engineer and manager of plant, Daimler Motoren Gesellschaft, *Stuttgart-Unterturkheim, Germany.*

VOORHEES, H. R., assistant engineer, United Motors Products Co., *Grand Rapids, Mich.*

VOSHALL, ROY D., superintendent of equipment, Washington Railway & Electric Co., *City of Washington.*

WATTER, MICHAEL, aeronautical engineer, Chance Vought Corporation, *Long Island City, N. Y.*

WEBER, ALBERT, general superintendent of foundry, Wilson Foundry & Machine Co., *Pontiac, Mich.*

WEBER, MAXIMILIAN, proprietor, Motor Car Service Co., *Chicago.*

WERTZLER, M. C., automotive engineer, Lincoln Co. Garage, *Caliente, Nev.*

WILLIAMSON, JOHN D., district sales manager, S. K. F. Industries, Inc., *New York City.*

WOBROCK, MARVIN T., draftsman, Hupp Motor Car Corporation, *Detroit.*

WOLCOTT, CARL H., draftsman, International Harvester Co., *Springfield, Ohio.*

WOLCOTT, KENNETH O., assistant general manager, North East Service, Inc., *Rochester, N. Y.*

WOODWARD, L. L., president and general manager, Fitzgibbon & Crisp, Inc., *Trenton, N. J.*

WOODWORTH, ERNEST H., tester, Boston Overland Co., *Boston.*

WYANT, I. A., secretary and treasurer, Campbell, Wyant & Cannon Foundry Co., *Muskegon, Mich.*

YOUNG, FENTON R., service department, Locomobile Co. of America, Inc., *Bridgeport, Conn.*

Applicants *Qualified*

The following applicants have qualified for admission to the Society between Jan. 11 and Feb. 10, 1926. The various grades of membership are indicated by (M) Member; (A) Associate Member; (J) Junior; (Aff) Affiliate; (S M) Service Member; (F M) Foreign Member.

ANDERSON, CARL A. (A) sales engineer, Hercules Motors Corporation, Canton, Ohio; (mail) 3047 Hull Avenue, *New York City.*

ARMSTRONG, W. F. (M) production manager, Olds Motor Works, *Lansing, Mich.;* (mail) Porter Apartments.

BACON, ELDRIDGE F. (J) 30 Lawn Street, *Dayton, Ohio.*

BERNARD, IRVING H. (A) manager of automobile sales, Wills Sainte Claire & Rickenbacker Sales, Burgan Building, 25 Martin Avenue, *White Plains, N. Y.*

BLAKEY, DONALD GREY (A) consulting engineer and automobile claims assessor, 7 Victoria Chambers, *Leeds, England.*

BOMAR, CAPT. ERNEST C. (S M) Ordnance Department, Watertown Arsenal, *Watertown, Mass.*

BOURQUE, WILLIAM D. (J) detailer, American Body Co., *Buffalo;* (mail) 59 Page Street.

BRAND, HUGO (A) chief engineer, Ambi-Werke, Berlin-Johannisthal, Germany; (mail) Bruckenstrasse 31, *Berlin-Niederschoeneweide, Germany.*

BRUNN, HERMANN A. (M) president, Brunn & Co., 980 Ellicott Street, *Buffalo.*

BUNDY, HARRY W. (A) president and general manager, Bundy Tubing Co., 4845 Bellevue Avenue, *Detroit.*

CALLAHAN, R. C. (J) research assistant, General Motors Corporation Research Laboratories, Building No. 9, *Detroit.*

CANTY, CAPT. DANIEL JOSEPH (S M) American graves registration service in Europe, 20 Rue Molitor, *Paris* (16), *France.*

CAPSEY, B. H. (A) general parts and service manager, Willys-Overland Sales Co., Ltd., Weston Road South, *Toronto, Ont., Canada.*

CASSON, WESLEY C. (J) designer, Cadillac Motor Co., Detroit; (mail) 72 Richton Avenue, *Highland Park, Detroit.*

COLE, PHILIP G. (A) president, A. Schrader's Son, Inc., 470 Vanderbilt Avenue, *Brooklyn, N. Y.*

CONNIFF, JOSEPH (M) superintendent of equipment, People's Motorbus Co. of St. Louis, 3615 Forest Park Boulevard, *St. Louis.*

COSTIGAN, WESLEY S. (A) works manager, Reynolds Spring Co., *Jackson, Mich.;* (mail) 754 Union Street.

COX, DONALD S. (M) research engineer, Pierce-Arrow Motor Car Co., 1695 Elmwood Avenue, *Buffalo.*

CRARY, RALPH W. (A) president, Nite-Eye Reflector Co., *Waukesha, Wis.;* (mail) P. O. Box No. 72.

CRAWFORD, FREDERICK C. (M) general manager, Steel Products Co., 7731 Conant Avenue, *Detroit.*

CROSBY, GEORGE A. (J) designing engineer, Brooks Steam Motors, Ltd., *Stratford, Ont., Canada;* (mail) 238 Cobourg Street.

CUTTING, HEYWARD (A) accessory manufacturer, Windsor Motor Products Co., *New York City;* (mail) 542 Fifth Avenue.

DEBBINK, HENRY L. (M) superintendent of gasoline vehicles, Milwaukee Electric Railway & Light Co., *Milwaukee;* (mail) 1001 44th Street.

DE HART, HENRY T. (A) advertising manager, Reo Motor Car Co., *Lansing, Mich.;* (mail) 528 West Kalamazoo Street.

DINSLAGE, FRANZ (F M) manager, Deutsche Kraftfahrzeugwerke A. G., Spandau, Germany; (mail) Marburger Strasse 5, *Charlottenburg, Germany.*

DIXON, J. F. (A) carbureter agent, 1835 South Hope Street, *Los Angeles.*

DRAKE, MAJOR CHARLES C. (S M) supervision of supply and transportation, Philippine Division Staff, Fort William McKinley, *Rizal, P. I.*

DUGAN, RAYMOND C. (A) Oldsmobile-Pittsburgh Co., 424 North Craig Street, *Pittsburgh.*

ERNST, PAUL J. (J) designer, International Motor Co., New Brunswick, N. J.; (mail) 105 Fulton Street, *Rahway, N. J.*

GETSINGER, R. C. (A) manufacturers' representative, Getsinger-Fox Co., 3-154 General Motors Building, *Detroit.*

GLITZENHIRN, GEORGE (A) instructor in shop practice, University of Michigan, *Ann Arbor, Mich.;* (mail) 121 West William Street.

GREEN, KANE S. (A) president, Automobile Club of Philadelphia, 23 South 23rd Street, *Philadelphia.*

GUINNESS, W. F. (J) draftsman, International Motor Co., New York City; (mail) 309 Marlborough Road, *Brooklyn, N. Y.*

HANNEWALD, BURTON (J) mechanical engineer, Charles D. Schmidt Corporation, New York City; (mail) 312 83rd Street, *Brooklyn, N. Y.*

HARROP, THOMAS NEWTON (F M) Osberton Radiators, Ltd., *Summertown, Oxford, England.*

HICKS, GEORGE (J) chassis draftsman, Six Wheel Co., *Philadelphia;* (mail) 2552 North Marshall Street.

HILL, JOSEPH B. (A) sales engineer, Ensign Carburetor Co., *Los Angeles;* (mail) 837 South Burlington Street.

HOLM, R. M. (A) sales promoter of automotive oils, Standard Oil Co. of Indiana, Chicago; (mail) 508 North Street, *Ishpeming, Mich.*

HOLTON, LEONARD (A) planning engineer, Murray Body Corporation, Detroit; (mail) 11 Farrand Avenue, *Highland Park, Detroit.*

HOWGRAVE, A. A. (A) manager for Upper Burma, G. McKenzie & Co.; (mail) c/o A. Scott & Co., *Mandalay, Burma, India.*

HYDE, DOUGLAS ANDERSON (J) office of vice-president and general manager, Fifth Avenue Coach Co., 605 West 132nd Street, *New York City.*

JAYES, HERBERT (A) designer of machinery, Crescent Washing Machine Co., New Rochelle, N. Y.; (mail) 52 Midland Avenue, *Rye, N. Y.*

JOHNSON, HAROLD L. (J) mechanical draftsman and checker, Sheldon Axle & Spring Co., *Wilkes-Barre, Pa.;* (mail) 455 Hazle Avenue.

JUHASZ, JOHN (M) president and general manager, Juhasz Carburetor Corporation, 244 West 49th Street, *New York City.*

KAISER, HERMAN (J) designing and layout draftsman, Federal Motor Truck Co., *Detroit;* (mail) 11762 Goodwin Avenue.

KILLIAN, FRANK B. (M) chief of engineering division, automotive department, Vacuum Oil Co., 61 Broadway, *New York City.*

KRAMER, EILEF F. (J) assistant engineer, Available Truck Co., *Chicago;* (mail) 3020 North Neenah Avenue.

KREIS, O. C. (M) chief research engineer, Continental Motors Corporation, *Detroit;* (mail) 2246 West Grand Boulevard.

LAZELLE, U. ELLIS (A) engineer, manufacturer's service division, Vacuum Oil Co., *Minneapolis;* (mail) 3233 Park Avenue.

LEA, HARRY S. (A) labor administrator, Yellow Truck & Coach Mfg. Co., 5801 West Dickens Avenue, *Chicago.*

LEMMER, CHARLES G. (A) sales engineer, Steel Products Co., *Detroit;* (mail) 363 Eastlawn Avenue.

LOEB, LUDWIG (A) director, Ambi-Werke, Berlin-Johannisthal, Germany; (mail) Humboldstrasse 26, *Berlin-Grunewald, Germany.*

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